IN TWO SECTIONS—SECTION ONE

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Requirements for Relief of Overpressure in Vessels Exposed to Fire

By J. J. DUGGAN,[1] C. H. GILMOUR,[1] and P. F. FISHER[1]

Pressure-vessel codes and regulations prescribe pressure-relief equipment but do not completely specify the necessary relief capacity to insure safety. Investigations show that the most effective cause of pressure increase is fire exposure as encountered in accidental conflagrations, and tests of the effects of such fire exposure on pressure vessels have been made and analyzed. Part 1 of the paper details these tests and analyzes the results. Parts 2 and 3 develop the necessary formulas to determine the sizes and capacities of the relief connections and apparatus and also present detailed statements of the applications of these formulas to pressure vessels and atmospheric tanks.

PART 1 OBSERVED RATE OF HEAT ABSORPTION

Introduction

FROM the standpoint of protection from excessive internal pressure, probably the severest hazard to which a vessel may be subjected is that which accompanies exposure to external conflagration. (Internal reaction will be discussed in Part 2.)

Some 12 to 14 years ago, the American Petroleum Institute Committee on Fire Prevention, after study and deliberation, proposed the basis of 100 Btu per min per sq ft of surface wetted by the contents as a practical basis for determining the necessary relief area to limit the rise of pressure. In connection therewith a theoretical analysis of "The Rates of Vaporization in Gasoline in Storage Tanks Exposed to Fire,"[2] and a chart from which to determine the capacity and size of relief were made. This chart was drawn so that heat-input rates varying from 6000 Btu to 24,000 Btu per sq ft of wetted surface per hr might be selected. So far as is known, there was no practical basis for the choice of a heat rate, nor was any suggested.

The National Fire Protection Association and the National Board of Fire Underwriters, as well as other regulatory bodies, were given the benefit of these studies. They performed a valuable service in this matter of emergency relief and related considerations by providing the best generally known criterion for the "Safe Handling and Storage of Flammable Liquids."[3] The relief standards proposed were based upon 6000 Btu per hr per sq ft of wetted surface and are still generally recognized.

Some interest appeared later among the associations and concerns promulgating and using this information to inaugurate a variable heat-input rate. It is believed the suggestion which received widest acceptance was that which proposed a heat rate varying from 24,000 Btu per hr per sq ft for small tanks having 10 sq ft of surface to 3000 Btu per hr per sq ft for large vessels having 10,000 sq ft of surface. On this basis, a 6000-Btu rate held for vessels having a wetted surface of 1000 sq ft. So far as is known, no official action was taken on this variable rate, nor was it published for general use.

It was the unfortunate experience of the authors to learn that, while the 6000-Btu rate on tank surfaces of the order of 1000 sq ft was satisfactory for all ordinary circumstances, it was totally inadequate for severe exposures of even less than 15 min duration. Some of these cases are a matter of private record, and it may be stated that the materials contained were more stable than gasoline upon which existing standards are based. From information reviewed, it is believed that other organizations have encountered similar experiences.

Accordingly, known available literature on the subject was analyzed and a series of tests was conducted which follows:

Summary

It is concluded that pressure-relief areas should be designed on a basis of 20,000 Btu per hr per sq ft of wetted surface exposed to fire.

Exception is proposed to this heat-input rate for atmospheric-working-pressure tanks, as outlined in Part 3 of this paper.

So far as can be learned, there is no substantial theory or evidence to support the limiting of heat-input rates to any particular height. There are some indications to the contrary; that is, high gas temperatures apparently exist at relatively great heights above the fuel during large fires.

Analysis of Fetterly's Formula and Test

In order to specify safety valves for the protection of containers charged with liquefied gas, a formula was derived and a test conducted by John F. Fetterly, inspector, Bureau of Explosives, Association of American Railroads. His report[4] furnishes data of considerable interest on heat-input rates for tanks exposed to fire.

It will be seen that it is possible for the contents of a 300-gal tank exposed to fire to absorb heat at about 20,000 to 23,000 Btu per sq ft of wetted surface per hr during vaporization.

Probably the factor in the formula which might arouse criticism is that represented by C, namely, the over-all heat-transfer coefficient. The coefficient used was obtained from an empirical formula which was based on tests made before the film theory of heat transfer was recognized. However, having been based on actual test data, it is not surprising that the numerical value of C is of the order of magnitude that would be predicted by present-day theory.

For example, the heat-transfer coefficient from a flame to a surface would be calculated today approximately as follows

$$U = h_c + h_r = \text{Fetterly's } C \text{ or heat-transfer coefficient} \quad [1]$$

$$h_c = 0.27\,(T_f - T_s)^{1/4} \quad [2]^{5}$$

[1] Carbide and Carbon Chemicals Corporation, South Charleston, West Va., unit of Union Carbide and Carbon Corporation.

[2] Engineering Office of American Petroleum Institute, New York, N. Y.

[3] National Fire Protection Association Flammable Liquids Ordinance, National Board of Fire Underwriters, Pamphlet No. 30 of Standards.

Contributed by the Petroleum Division and presented at the Semi-Annual Meeting, Los Angeles, Calif., June 14–17, 1943, of The American Society of Mechanical Engineers.

[4] "The Determination of the Relief Dimensions for Safety Valves on Containers in Which Liquefied Gas Is Charged and When the Exterior Surface of the Container Is Exposed to a Temperature of 1200 Deg F," by J. F. Fetterly, Bureau of Explosives, 1928.

[5] "Heat Transmission," by W. H. McAdams, McGraw-Hill Book Company, Inc., New York, N. Y., 1942, second edition, Equation [16], p. 240.

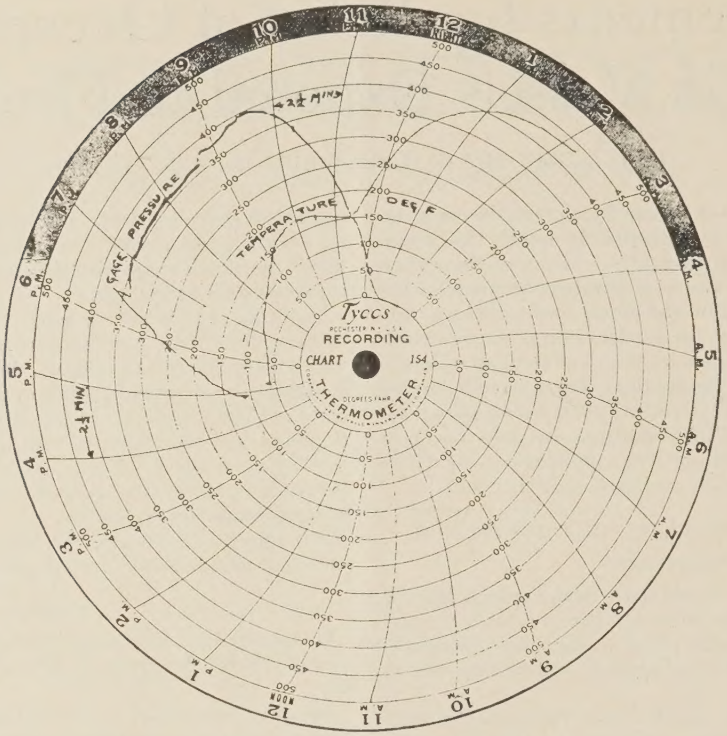


Fig. 1 Fetterly's Tank Fire-Exposure Test[4]

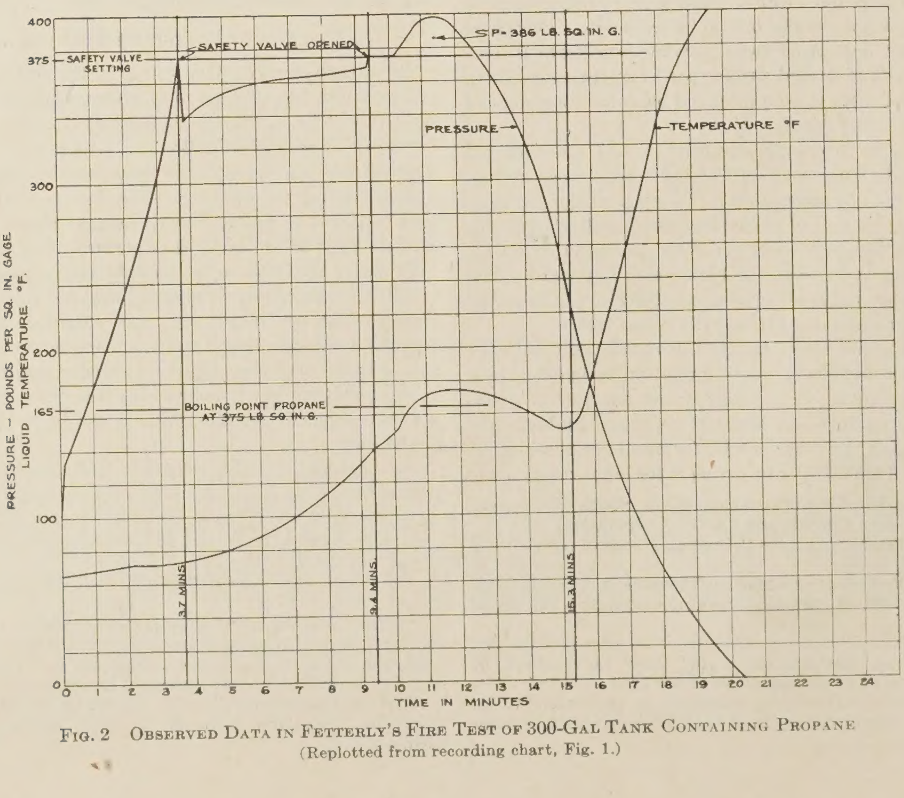


Fig. 2 Observed Data in Fetterly's Fire Test of 300-Gal Tank Containing Propane
(Replotted from recording chart, Fig. 1.)

$$h_r = \frac{0.173\left[\left(\frac{T_f}{100}\right)^4 - \left(\frac{T_s}{100}\right)^4\right]}{(T_f - T_s)} \qquad [3]^6$$

(Assuming the hot gas and tank surface are black, i.e., have a black-body coefficient of 1, the shape factor is unity, and that the heat resistance of the retaining wall and boiling liquid is negligible compared with the resistance of the film in contact with the flame.) Then if the flame is 1200 F and tank surface is 165 which are the temperatures used by Fetterly, the value of U would be

$$U = 0.27\,(1660 - 625)^{1/4} + \frac{0.173[(16.6)^4 - (6.25)^4]}{(1660 - 625)}$$

$$= 0.27 \times 5.67 + \frac{0.173\,(75{,}940 - 1540)}{1035}$$

$$= 1.53 + 12.44 = 13.97 \text{ Btu per hr per sq ft per deg F}$$

This is more than 95 per cent of the value obtained by Fetterly, specifically 14.68 Btu. It should be noted that this is equivalent to [13.97 (1200 — 165)] 14,500 Btu per hr per sq ft of wetted surface for a flame temperature of 1200 F, and a 23,000-Btu rate for a T_f of the order of 1400 F, as calculated from the test data to be given. It will be seen later that flame temperatures of this magnitude are conservative for industrial fires, particularly those involving flammable liquids.

Of particular interest and value are the test data obtained by Fetterly which consist of continuous pressure and temperature measurements taken while a wood fire burned around a tank containing 1000 lb of propane. From this record, it is possible to calculate the heat input to the tank and contents during the various phases of the test. The charts of pressure and temperature versus time are reproduced in Figs. 1 and 2 of this paper; the former is the actual instrument record and the latter is a replot of the same data to clarify the analysis. In order to give a better picture of the nature of the fire surrounding the tank, a diagram of the test setup, as interpreted from the description contained in Fetterly's paper, is shown in Fig. 3.

The safety valve was set to open at 375 psig. Note that the relief valve opened and closed again long before the liquid approached the boiling point (about 165 F), which is evidence that the vapor was superheated to a temperature sufficient to cause the pressure to rise to 375 psi. By reference to Fig. 3, it is obvious that the vapor space was subjected to intense fire for the first few minutes of the test, during which time the vapor was heated from 65 F to an estimated temperature of 1000 F, at the time the safety valve first opened after 3.7 min of exposure. This temperature was estimated from the best available physical characteristics[7] for propane. In this same period the liquid was heated from only 65 to 74 F.

The second phase of the test consists of the period in which part of the liquid was heated to the boiling point and the safety valve opened for the second time. This is the interval observed from 3.7 to 9.4 min in Fig. 2. During this time the fire level probably lowered, the intensity of the flame on the unwetted surface decreased, and the vapor did not expand as rapidly as before, as observed from the pressure curve.

The final phase of the test consists of the period of the vaporization and discharge of the contents. As shown in Fig. 2, this appears to have occurred in about 5.9 min. At the end of this period, the pressure had fallen rapidly to 245 psi (probably the safety valve stuck open), and the liquid temperature began to rise sharply as it would when the thermocouple was exposed to the hot vapor and tank walls. On the basis of 5.9 min, the heat absorbed by the contents may be calculated as follows:

Heating Liquid to Boiling

$$1000 \times 0.68\,(165 - 140) \times \frac{60}{3.2} = 318{,}800 \text{ Btu per hr}$$

Vaporizing Liquid

$$1000 \times 101 \times \frac{60}{5.9} = \frac{1{,}027{,}000 \text{ Btu per hr}}{1{,}345{,}800 \text{ Btu per hr}}$$

Heat Intensity

$$\frac{1{,}345{,}800}{57.82} = 23{,}300 \text{ Btu per hr per sq ft of wetted surface}$$

Vaporization Rate

$$1000 \times \frac{60}{5.9} = 10{,}170 \text{ lb per hr}$$

Safety-Valve Capacity

$$W = 306\,aP\sqrt{\frac{M}{T}} = 306 \times 0.312 \times 386\sqrt{\frac{44}{625}}$$

$$= 9800 \text{ lb per hr} \qquad [4]^8$$

The vaporization rate is then 3.8 per cent greater than the calculated valve capacity, but some of the contents may have been discharged as droplets of liquid. However, if the vaporization rate had been as low as the safety-valve capacity, the heat intensity would still have remained as high as 22,600 Btu per hr per sq ft of wetted surface.

If it is assumed that the steel temperature is about 100 deg F higher than that of the boiling liquid, the effective flame temperature necessary to transmit the foregoing heat input may be calculated as follows:

$$(1) \quad U\Delta T = Q/A\theta = U(T_f - T_s) = 23{,}300 \text{ Btu per hr per sq ft} \qquad [5]$$

$$(2) \quad U = 0.27(T_f - T_s)^{1/4} + \frac{0.173\left[\left(\frac{T_f}{100}\right)^4 - \left(\frac{T_s}{100}\right)^4\right]}{(T_f - T_s)} \qquad [1]$$

[6] "Heat Transmission,"[5] Equation [17], p. 63.

[7] Pressure-volume-temperature relation.

[8] "API-ASME Code for Unfired Pressure Vessels," New York, N. Y., 1938 edition, p. 74.

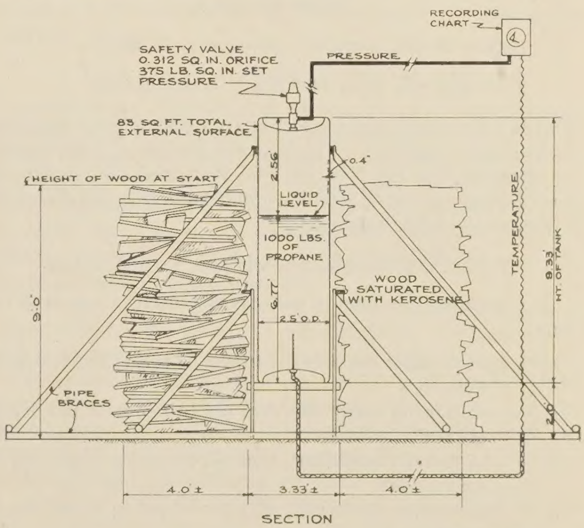


Fig. 3 Diagrammatic Representation of the Fire-Exposure Test of a 300-Gal Tank Containing Propane

(As described in ref. 4.)

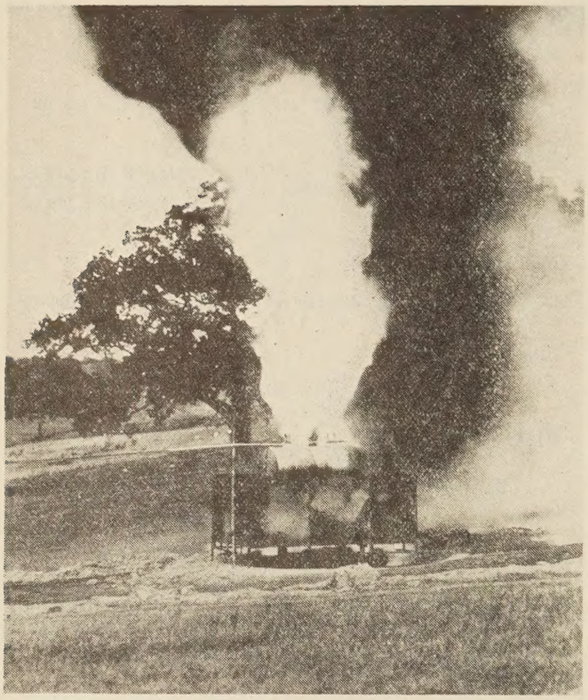


Fig. 4 Fire Conditions 9 Min After Ignition
(Appears to be height of fire, but note limited surface exposure.)

$$U(T_f - T_s) = 0.27\,(T_f - T_s)^{5/4} + 0.173\left[\left(\frac{T_f}{100}\right)^4 - \left(\frac{T_s}{100}\right)^4\right] = 23{,}300 \text{ Btu per hr} \tag{3}$$

$$T_s = 165 + 100 = 265 \text{ deg F} = 725 \text{ deg R} \tag{4}$$

$$0.27(1888 - 725)^{5/4} + 0.173[(18.88)^4 - (7.25)^4] = 23{,}300 \text{ Btu per hr} \tag{5}$$

$$1834 + 21{,}503 = 23{,}337 \text{ compared to } 23{,}300 \text{ Btu}$$

$$T_f = 1888 \text{ deg R, or } 1428 \text{ deg F, approximately}$$

This is a reasonable mean effective temperature for a kerosene-soaked open wood fire. The National Bureau of Standards in co-operation with the National Fire Protection Association observed temperatures of 2000 F within 20 min from burning wood within a large brick building.[9]

Nomenclature Used in Fetterly Test Analysis

The nomenclature used in analysis of Fetterly's test is as follows:

C = heat-transfer coefficient, Btu per hr per sq ft per deg F; Fetterly's
U = heat-transfer coefficient, Btu per hr per sq ft per deg F; over-all
h_c = heat-transfer coefficient, Btu per hr per sq ft per deg F; convection
h_r = heat-transfer coefficient, Btu per hr per sq ft per deg F; radiation
T_f = effective flame temperature, deg F absolute
T_s = temperature of tank surface, deg F absolute
ΔT = temperature difference, deg F
Q = total heat transferred, Btu
A = surface heated, sq ft
θ = time, hr
W = vapor rate of safety valve, lb per hr
a = orifice area of safety valve, sq in.
P = absolute pressure at inlet of safety valve, psi
M = molecular weight of vapor
T = vapor temperature, deg F absolute

[9] "Handbook of Fire Protection," by Crosby, Fiske, Forster, National Fire Protection Association, Boston, Mass., 1941, ninth edition, p. 384.

Analysis of Underwriters' Laboratories, Inc., Test

In January, 1938, there appeared the results of some tests to show the effect of a film of water running over a plate exposed to fire.[10] The results of the tests were reported in terms of temperature or per cent of total heat input. The data are comprehensive enough to be used to calculate the heat absorbed from the flame.

The test apparatus consisted of a vertical plate 8 ft high by 3 ft wide by $^1/_8$ in. thick suspended from one of its 3-ft edges. A fire was produced by burning gasoline in a pan 3 ft sq, placed directly in front of the plate. Tests were conducted both with and without a water film flowing down the plate.

After equilibrium had been established during the run, in which there was a protective water film, the following data were observed:

Water rate, 45 gpm; water temperature off, 97.5 F; water temperature on, 63 F; water temperature rise, 34.5 F.

From these data, the heat-absorption rate is calculated as follows:

$$45 \times 8.337 \times 60 \times 1 \times 34.5 = 776{,}000 \text{ Btu per hr}$$

[10] "Opacity of Water to Radiant Heat Energy," Underwriters' Laboratories, Inc., Bulletin of Research No. 3, Chicago, Ill., Jan., 1938.

Fig. 5 Conditions Interpreted to Be About 25 Min After Ignition
(Note no fire is visible in pan under tanks.)

The intensity of heat absorption is

$$\frac{776{,}000}{24} = 32{,}300 \text{ Btu per hr per sq ft}$$

The average temperature of the water was 80 F, when flowed over the plate during the fire. The effective gas temperature required to transmit this quantity of heat would be approximately 1580 F, as calculated by the same procedure as given for the analysis of the Fetterly tests. A maximum flame temperature, a point temperature, of 3000 F was read with an optical pyrometer.

The results of this test show that the intensity of heat absorbed from a fire under these conditions is of the order of magnitude of 30,000 Btu per hr per sq ft.

Fire Tests on Truck Tanks

During 1930, the Aluminum Company of America, under supervision of committees from the American Petroleum Institute and the National Fire Protection Association, conducted fire-exposure tests[11] on two 150-gal tank-truck compartments made of aluminum (refer to Figs. 4 and 5). The data were taken to establish the suitability of the metal for truck-tank construction and are not sufficiently broad to record directly information essential to the calculation of heat-input rates. As will be explained, sufficient data were interpreted to estimate maximum and minimum heat-input rates for tank B in the report.

The time-temperature curves of thermocouples 9, 10, 11, and 12 were transposed from the original data[11] to Fig. 6 of this paper. This indicates by the abrupt temperature rises that the gasoline between three fourths and one fourth of the tank height vaporized in 10 min. By using the dimensions in the report, this was calculated to have been 90 gal. It appeared from the illustrations in the report that probably an area equivalent to not more than the lower half of the tank less one head was exposed to the flame during the period. This is about 15 sq ft of area. The gasoline weighed 6.3 lb per gal at normal conditions. Its latent heat of vaporization was about 130 Btu per lb. Then the heat absorbed would be calculated as follows:

$$\frac{90 \times 6.3 \times 130}{15} \times \frac{60}{10} = 29{,}500 \text{ Btu per hr per sq ft}$$

The data indicate that the exposure fire lasted in all about 25 min, during which time the tank metal was heated, the gasoline raised to the boiling point, and 129 gal were vaporized. On this over-all basis, the intensity of the fire to the tank may be calculated, as follows:

To Heat Metal

113 lb × 0.23 × 325 deg avg *dt* = 8,450 Btu

To Heat Liquid to Boiling Point

147 gal × 6.3 lb × 0.57 × 207 deg avg *dt* = 109,200 Btu

Latent Heat of Vaporization

129 gal × 6.3 lb × 130 = 105,700 Btu

223,350 Btu

Intensity

$$\frac{223{,}350}{31.5 \text{ sq ft total exposed}} \times \frac{60}{25} = 17{,}000 \text{ Btu per hr per sq ft}$$

The time-pressure curves show that rapid ebullition started at about 4 min after ignition. The vaporization period for 129 gal was then about 21 min, and a heat-input rate to the contents may be approximated:

$$\frac{105{,}700}{15} \times \frac{60}{21} = 20{,}100 \text{ Btu per hr per sq ft.}$$

It is concluded that the actual maximum heat-input rate lies between 20,000 and 29,000 Btu per hr per sq ft of wetted surface exposed.

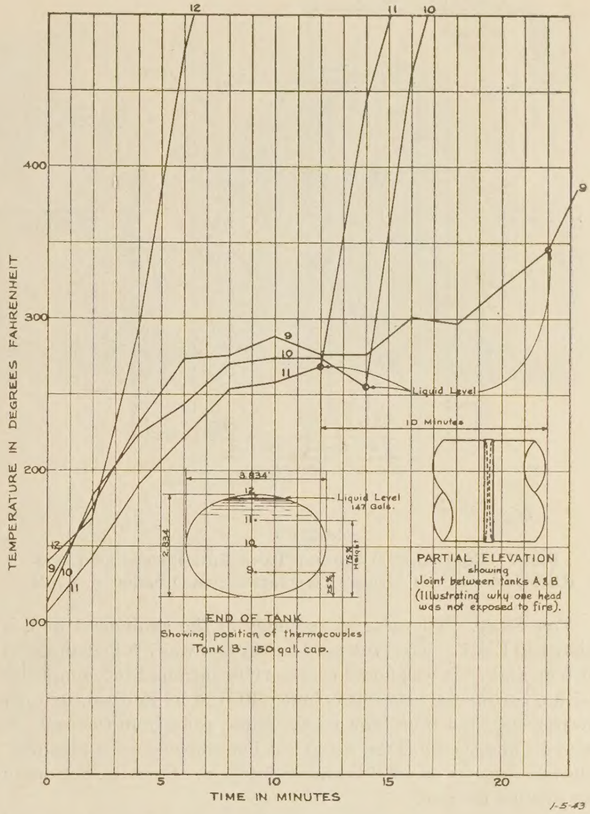


Fig. 6 Time-Temperature Curves
(Transposed from Fig. 4 of ref. 11.)

Tank Fire-Exposure Tests by the Authors

Introduction. Since existing information on the subject seemed inadequate, a series of exposure tests was conducted by the authors' company, in which a 3000-gal nominal-capacity tank containing 2300 gal of water was surrounded by an intense fire. These were begun in the fall of the year 1938 and were concluded in April, 1940. Herein are described and analyzed the last four tests, since the results obtained during these are considered the more accurate and comprehensive.

The tests were conducted (1) to establish a safe minimum heat-input rate for the basis of requirements for emergency relief, (2) to determine the necessary rate of application of external cooling-water film both to maintain a specified temperature within the contents and to approximate the amount required to preserve such vessels when involved in a conflagration. The latter consideration is the subject of a separate discussion, the present purpose being to determine a heat-input rate.

Summary. When a tank is surrounded by an intense fire having an effective temperature of about 1400 F, heat is absorbed from the flame at a rate of the order of 20,000 Btu per hr per sq ft of wetted surface exposed.

[11] Report on "Impact, Hydrostatic, and Fire Tests—Aluminum Alloy Compartments for Tank Trucks," Aluminum Company of America, New Kensington, Pa., 1930.

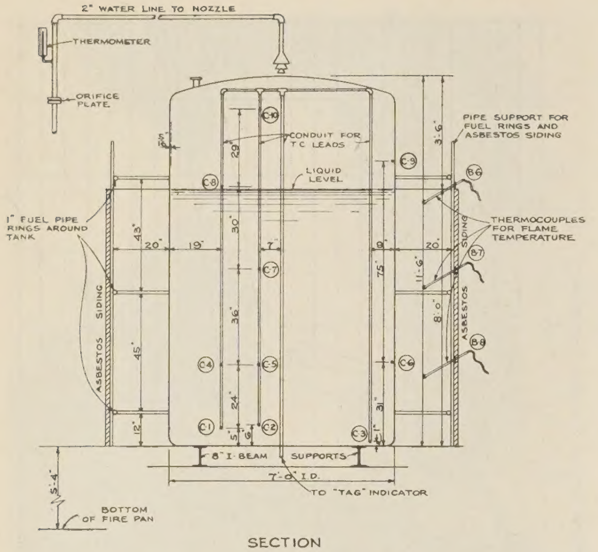


Fig. 7 Equipment Assembly Tank Fire-Exposure Tests
(Carbide & Carbon Chemicals Corporation, South Charleston, W. Va.)

Test Setup. The general assembly of the equipment used is shown in Fig. 7. The tank used for the test was 7 ft in diam × 11 ft 6 in. high. It was found necessary to surround the equipment with a corrugated asbestos cylinder 10 ft, 4 in. in diam, in order to prevent wind from blowing the flame away from the tank, as may be observed in Figs. 8 and 9. For convenience and safety, the tank was set on steel beams at 5 ft 4 in. above the bottom of an existing fire pan.

The temperatures were measured with thermocouples placed inside the tank, on the tank wall, and in the flame. A direct-reading "Celectray" potentiometer was used for converting the electromotive-force readings to deg C. Mercury-in-glass thermometers were used to measure the cooling water and atmospheric temperatures.

The cooling water was measured with an orifice and was released on top of the tank through a funnel-type nozzle. It flowed over the head and wall in a uniform film and was collected in a trough. The cooling water was not only a part of the experiment but preserved the vessel as intended.

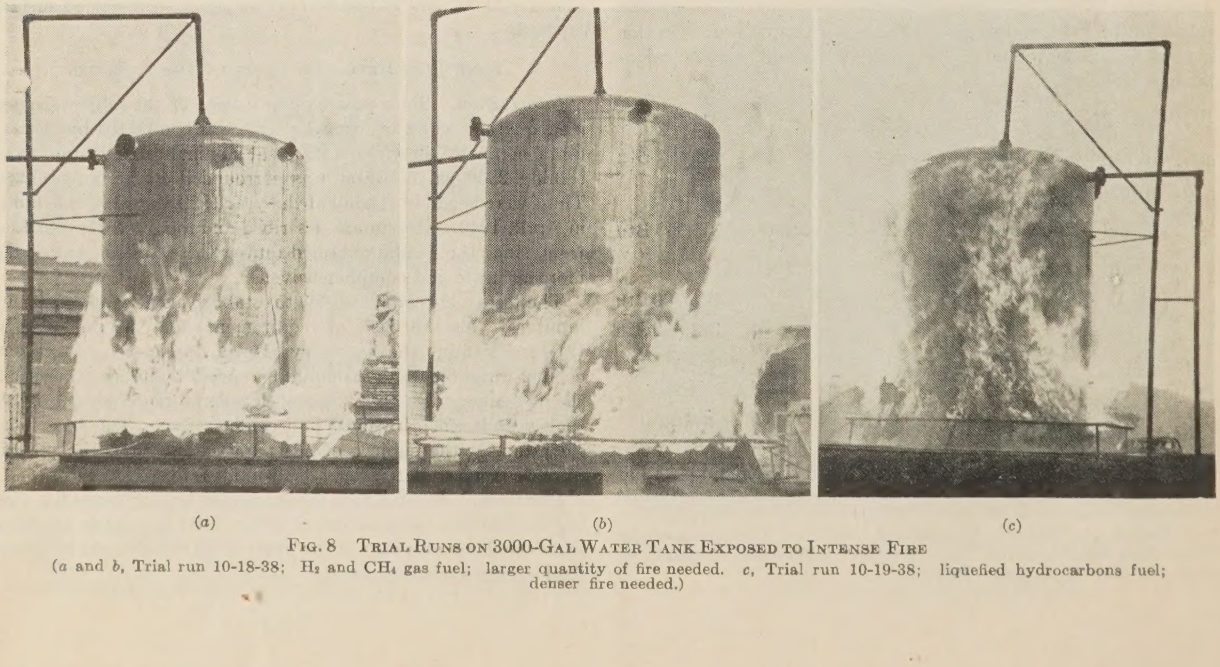
The fire was made by burning hydrocarbons in atmospheric air. The fuel was released through three 1-in. standard-pipe perforated rings, concentric with the tank. The fuel rings were about 4 ft apart vertically, as shown in Fig. 9, and from inspection produced a flame which surrounded the tank thoroughly. The liquid fuel was measured by observed difference in level, as indicated by calibrated tank gages.

Test Procedure. At the start of the tests, the tank contained 2300 gal of water. After making zero readings of the temperatures, the cooling water was started and the fuel was ignited. The fuel rate was kept relatively constant throughout a test run. The temperatures of the flame, cooling water, atmosphere, and contents (water) were recorded at approximately 15-min intervals. The cooling-water rate was adjusted to three values, 143 gpm, 71 gpm, and 37 gpm. No cooling water was used during run No. 4 and partial boiling occurred. During this last test, the fuel rate was reduced to less than one third that of the previous runs, but the flame temperature at the points read was maintained.

Data Obtained. When the temperature of the contained water remained constant, a condition of equilibrium existed, and the data shown in Table 1 correspond to that phase of the first three runs. During test No. 4 in which no cooling water was used, the experiment was stopped before stabilized conditions were reached (contents averaged 73.5 C), in order to avoid possible rupture of the vessel.

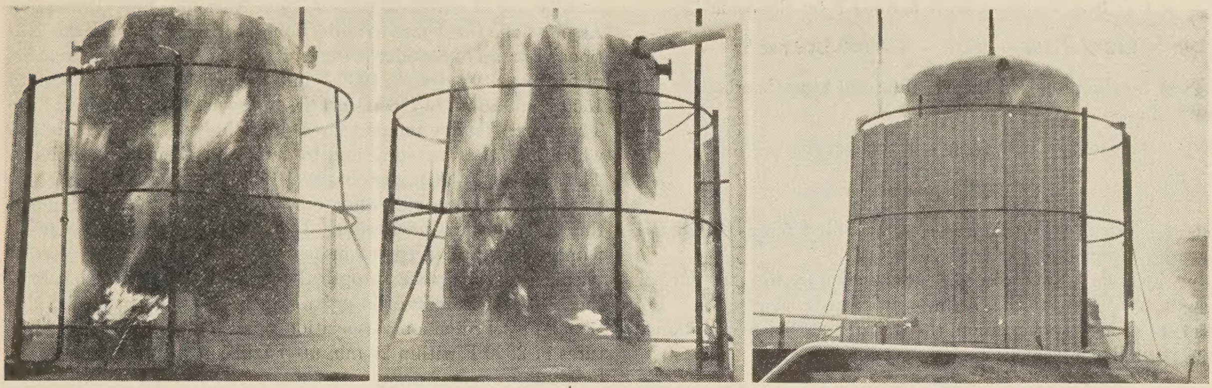
Discussion of Data. Surrounding the equipment with a cylinder to control windage no doubt increased radiation to the tank, but it is believed conditions are equally as severe where a vessel is surrounded by flame of considerable depth. As a counteractant to this effect, there occurred a strong updraft between the cylinder and tank. This was partly due to the 5-ft clearance under the tank as described.

The flame temperatures read are not believed to be maximum since the unshielded thermocouples could "see" the cooling-water film. The lower flame temperatures shown in runs Nos. 2 and 3 were apparently due to wind and rain. This resulted in lowered heat-input rates during these runs.



(*a*) (*b*) (*c*)

Fig. 8 Trial Runs on 3000-Gal Water Tank Exposed to Intense Fire
(*a* and *b*, Trial run 10-18-38; H_2 and CH_4 gas fuel; larger quantity of fire needed. *c*, Trial run 10-19-38; liquefied hydrocarbons fuel; denser fire needed.)



(a) (b) (c)

FIG. 9 FURTHER RUNS ON TANK FIRE-EXPOSURE TESTS

(Test runs 11-2-38; liquefied hydrocarbon fuel released through three rings of pipe; windshield in place.)

TABLE 1 OBSERVED AND CALCULATED DATA PERTAINING TO A 3000-GAL STEEL TANK CONTAINING 2300 GAL OF WATER AND EXPOSED TO A FIRE

	Run No. 1	Run No. 2	Run No. 3	Run No. 4
Total cylindrical surface exposed to fire, sq ft	242	242	242	242
Cylindrical surface in contact with contained liquid, and exposed to fire, sq ft	176	176	176	132[a]
Rate of cooling water on external surface of tank, gpm	143	71	37	0
Temperatures, average, deg C:				
Cooling water off	61.2	61.3	58.0	...
Cooling water on	12.5	12.2	11.5	...
Cooling water rise	48.7	49.1	46.5	...
Cooling water average	37.0	37.0	35.0	...
Flame temperature	820	690	714	721
Water inside tank	58	84	92	9 to 73.5 per hr
Surface of tank, external	58	84	92	12 to 100 per hr
Surface of tank, internal	58	84	92	10 to 95 per hr
Surface of tank not in contact with contained water	36	58	66	10 to 190 per hr
Surface of tank, log mean average	46	70	78	10 to 140 per hr
Fuel vapor rate, cfh	40,857	40,857	40,857	12,200
Total available heat in fuel, Btu per hr	106,000,000	106,000,000	106,000,000	29,900,000
Heat absorbed by tank, contents, and cooling water, Btu per hr	6,270,000	4,186,000	4,522,000	2,311,100
Heat absorbed by radiation, Btu per hr	5,760,000	3,737,000	4,050,000	2,100,000
Heat absorbed by convection, Btu per hr	510,000	449,000	472,000	211,100
Sensible heat absorbed by cooling water, Btu per hr	6,270,000	3,137,000	1,550,000	0
Latent heat absorbed by partial evaporation of cooling water, Btu per hr	0	1,049,000	2,972,000	0
Heat absorbed by tank contents, Btu per hr	0	0	0	2,225,000
Rate of heat absorption (heat density), Btu per hr per sq ft	25,900	17,300	18,700	16,850 by contents

[a] See section "Discussion of Data."

During run No. 4 in which no cooling water was used and the fuel rate was reduced to less than one third by using only the lower fuel-pipe ring, it was estimated that 2 ft of the wetted height of the tank was not exposed to flame. Conditions were similar to those shown in Fig. 8 (*a* and *b*). This reduced the wetted surface exposed to 132 sq ft.

The experiment covered three conditions of a tank exposed to fire: namely,

1 Contents maintained at a certain maximum temperature as in run No. 1. The vent rate was kept low and the absorbed heat appeared as sensible heat in the cooling water film.

2 Contents heated to higher maximum temperatures as in runs Nos. 2 and 3. The absorbed heat appeared as both sensible heat and heat of vaporization in the cooling-water.

3 Contents allowed to heat up as under actual conditions where a tank is unprotected and exposed to fire. The vent rate became high and the absorbed heat appeared as sensible heat in the contained water.

Observed Heat Absorbed Compared With Theoretical Calculations. The data indicate that the flame transmits heat to the tank by radiation and that the heat thus transmitted is again transferred by convection to the water film flowing over the tank surface. As a result, no heat is transmitted to the contents after equilibrium conditions are established. The film of water flowing over the surface of the tank also absorbed heat directly from the flame by convection, and under conditions of low water flow, as in runs Nos. 2 and 3, some water is vaporized from the flowing film. Calculations for this apparent mechanism of heat transfer compare well with the observed heat absorbed in runs Nos. 1, 2, and 3 and are submitted as further evidence to support the heat-input rates tabulated for these runs. The sensible heat absorbed during run No. 4 is direct evidence and, of course, not affected by this theory since no cooling water was used.

Calculated values for run No. 1 follow:

The sensible heat absorbed by the water film is

$$143 \times 8.337 \times 60 \times 1.0 \times 1.8 \times 48.7 = 6{,}270{,}000 \text{ Btu per hr}$$

or 25,900 Btu per hr per sq ft

Theoretically, the heat transmitted to the tank surface by radiation using the data in Table 1 is

$$h_r = 18.5 \text{ Btu per hr per sq ft per deg F by Equation [3]}$$

$$18.5 \times 242 \times (1508 - 114.8) = 6{,}240{,}000 \text{ Btu per hr}$$

The heat absorbed by the water film by convection using the data in Table 1 is

h_c = 1.65 Btu per hr per sq ft per deg F by Equation [2]

$$1.65 \times 242 \times (1508 - 98.6) = 563{,}000 \text{ Btu per hr}$$

Then by the proposed theory, the total heat absorbed by the water film is

$$6{,}240{,}000 + 563{,}000 = 6{,}803{,}000 \text{ Btu per hr}$$

or 28,100 Btu per hr per sq ft

This theoretical, total heat absorption is only 8.5 per cent greater than the observed heat absorbed.

By the theory, the radiated heat absorbed by the tank surface is again transferred to the cooling-water film by convection. For this type of heat transmission, the transfer coefficient is obtained from the following equation

$$h_w = \frac{1}{2} \cdot \frac{f^{2/3}}{(CZ/h)^{2/3}} C(2g\rho^2\Gamma)^{1/3} \ldots\ldots\ldots\ldots [6][12]$$

Using this and the observed data in Table 1, the heat absorbed from the tank surface by the water film is

h_w = 1490 Btu per hr per sq ft per deg F, by Equation [6]

$$1490 \times 242 \times (114.8 - 98.6) = 5{,}840{,}000 \text{ Btu per hr}$$

This is only 6.41 per cent less than the calculated heat absorbed by the tank surface by radiation as given (6,240,000) and apparently establishes the heat-transfer theory.

Further, according to theory $\left(\frac{18.5}{(18.5 + 1.65)} \times 100\right)$, 91.8 per cent of the heat absorbed by the cooling water was taken by convection from that radiated to the tank surface, so

$$6{,}270{,}000 \text{ (observed)} \times 0.918 = 5{,}760{,}000 \text{ Btu per hr}$$

This compares well with the 5,840,000 Btu per hr calculated with the theoretical heat-transfer coefficient (1490).

Calculations for run No. 4 are as follows:

The sensible heat absorbed by the contents water is

$$2300 \times 8.337 \times 1.0 \times 1.8\ (64.5) = 2{,}225{,}000 \text{ Btu per hr}$$

or 16,850 Btu per hr per sq ft

The observed heat absorbed by the tank is

$$242 \times \frac{0.3125}{12} \times 487 \times 0.12 \times 1.8\ (130) = 86{,}100 \text{ Btu per hr}$$

Total heat absorbed.......................2,311,100 Btu per hr

This observed heat absorbed is only 8.45 per cent less than the theoretical heat input when calculated as for run No. 1. The calculations indicate the heat absorbed by radiation to be 90.8 per cent of the total heat absorbed by the tank.

Flame Temperatures and Effective Height

Since the heat-input rate to a vessel exposed to fire is sensitive to the mean effective flame temperature, as previously noted under the discussion of Fetterly's work, it is well to emphasize the fact that high temperatures are common during fires. It has been shown that effective flame temperatures of about 1400 F produce heat-input rates of the order of 20,000 Btu per hr per sq ft of wetted surface exposed.

The standard time-temperature curve,[13] used by the American Society for Testing Materials, the National Fire Protection Association, the Underwriters' Laboratories, Inc., and the National Bureau of Standards for a "standard scale of measurement of fire severity" rises to 1000 F in 5 min, to 1300 F in 10 min, to 1550 F in 30 min, to 1700 F in 1 hr, and so on up to 2300 F, in 8 hr. These are effective temperatures since they are furnace heats to which materials are subjected to determine their fire resistivity. The authors of this paper observed 1600 F temperatures with unshielded thermocouples while burning liquefied hydrocarbons in the open. The Underwriters' Laboratories, Inc., read 3000 F flame temperatures (point or maximum) from an open pan of burning gasoline with an optical pyrometer. As stated before, the National Bureau of Standards in co-operation with the National Fire Protection Association observed wood-fire temperatures of 2000 F within 20 min after ignition within a large brick building.

The authors are unable to learn of a substantial theory or to find convincing evidence to support the limiting of heat inputs into vessels exposed, to any particular height above the fuel. Insurance underwriters' recommendations of fireproofing supporting columns within 20 ft of a source of fire, as well as the practice by reliable organizations to limit heat inputs to 20 ft elevation, have been noted, but these do not seem adequate. Aluminum (1125 F to 1490 F melting point), glass (1400 F to 1600 F melting point), and brass (1575 F to 1800 F melting point) have been observed melted more than 20 ft above grade after flammable-liquid-spillage fires in open areas. These fires were of less than 1 hr duration. Examination of the analysis of fires by the National Fire Protection Association and the Factory Mutual Fire Insurance Companies, particularly those involving storage-tank farms and elevated tanks, discloses that high heats do prevail at comparatively great heights above the fuel during severe fires.

The foregoing evidence and experience support the following suggestion:

Where vessels are in confined areas, buildings, or enclosures, and an intense fire could occur, no reduction from the maximum height of the surface to be wetted by the contents should be used when designing emergency pressure-relief or other fire protection. Where such equipment is in open areas, properly spaced, diked, and otherwise installed in accordance with the insurance inspection organization having jurisdiction, it is considered not economical to design for a heat-input height of less than 40 ft, and unsafe under average conditions to use less than 30 ft. Further, no reduction of heat input is proposed (except for atmospheric tanks as outlined in Part 3) to allow for the possibility that vessels may not be completely surrounded with flame during a general fire.

Conclusion

A rate of heat absorption for vessels exposed to fire as a basis of requirements for relief of overpressure is concluded as outlined under Summary from the foregoing experiments, theory, and experience.

The employment of this basis follows in Parts 2 and 3 of this paper.

The authors are cognizant that the observed absorption rate (20,000 Btu per sq ft per hr) appears high when compared to overall heat-absorption rates of boilers, tube stills, etc. It is reasonable, however, when compared to those rates[14] for waterwall sec-

[12] "Principles of Chemical Engineering," by Walker, Lewis, McAdams, and Gilliland, McGraw-Hill Book Company, Inc., New York, N. Y., 1937, Equation [11c], p. 114.

[13] "Handbook of Fire Protection,"[9] p. 380.

[14] For further information on absorption rates in boilers, refer to "Kent's Mechanical Engineers' Handbook," vol. 2, "Power," John Wiley and Sons, Inc., New York, N. Y., eleventh edition, 1937, pp. 6–17, 6–18, and 6–36; "The Trend of Boiler Design," Bulletin 3-180, The Babcock and Wilcox Company, New York, N. Y., 1935, pp. 3 and 4; "The Simple and Direct Answer to All Problems of Heat Transfer in Boilers," by L. R. Stowe, Wilmette, Ill., 1937, chapters 12 and 13.

tions and tubes adjoining combustion chambers which may absorb heat at rates from 50,000 to 150,000 Btu per sq ft per hr, based upon the projected area.

PART 2 REQUIREMENTS FOR RELIEF OF PRESSURE VESSELS[15]

Introduction

One of the fundamental considerations in all codes and regulations applying to boilers and unfired pressure vessels has been that of preventing overpressure and its attendant hazards. In all boiler codes and regulations, it has been almost universal practice to base the size and capacity requirements for safety valves on the potential heat input, which is computed from the total area of the heat-absorbing surfaces of a boiler. In codes and regulations applying to unfired pressure vessels, however, while pressure-relief devices are required and an overpressure limit is specified, there is no measurable basis upon which to compute the capacity of such relief devices. In Paragraph U-2 of the A.S.M.E. Unfired Pressure Vessel Code,[16] the question of relieving capacity is referred to by specifying that the relief devices shall be of such capacity as to prevent a rise of more than 10 per cent above the maximum allowable working pressure, but the user of a pressure vessel receives no directions from that code as to how to proceed to compute that capacity.

Actual operating experiences in industrial plants tend to give the general impression that an appreciable number of "code pressure vessels," if investigated, would be found to be underequipped with pressure-relief capacity. It is often contended that pressure vessels equipped with automatic controls are inherently more safe than those not so equipped; yet it is a matter of record that when such automatic controls fail or become inoperative (as will happen occasionally) an accident often follows. The very natural relaxation of vigilance, which comes from depending upon automatic controls, causes neglect of the pressure-relief devices; and if they will operate, they are likely to be found inadequate in capacity. When pressure vessels do fail in service, it is always found that some one of the important safety considerations has been overlooked or neglected, and in many of such cases, it is found that the pressure-relief devices are inadequate in capacity.

Our experience during the past 8 years, in inspecting thousands of pressure vessels and testing thousands of relief valves and safety devices, has indicated a very definite need for more information on the subject of pressure relief, and the authors have undertaken an investigation to obtain additional data. The results to date appear to warrant presentation of a statement in order to learn whether the data will be useful to the A.S.M.E. Code Committee and other code-making bodies. The authors have in the past found the Unfired Pressure Vessel Code a most valuable aid in their work and, in appreciation thereof, desire to offer their cooperation in every way possible. If the data obtained appear likely to prove helpful to the code committees, they will be placed at the disposal of such committees, and if necessary, additional information will be gathered.

In any comprehensive study of this broad question of protection against overpressure, it is advantageous to consider the sources from which pressure is induced in vessels. In a large percentage of cases, the source of the pressure is mechanical pumping or compression with pressure regulation possible at the source, as is exemplified in air compressing and refrigerating systems; in these, overpressure is likely only on the rare occasions when control apparatus fails. In the large number of cases where pressure is transmitted from the source through systems of piping, as in chemical and petroleum processes, there is no such simplified control possible; and the pressure may be increased from overfilling, overheating, chemical action, or improper operation. Of these several causes, probably the most serious is overheating. In chemical and petroleum plants, where the condition of volatile contents in vessels is inherent and in cases where pressure vessels with relatively nonvolatile contents have been subjected to exposure fires, disastrous explosions have occurred. This class of equipment (chemical and petroleum) is probably representative of those most difficult to control from a safety viewpoint.

It is the observed tendency of codes and regulations to call for safety protection to meet the worst possible hazard which may be encountered and which, in pressure-vessel operation, is probably the overheating that may result from exterior fire exposure. In the past, it has been customary (with few exceptions) to treat such disasters as acts of the elements which are beyond human control, but our investigations tend to show that this conclusion is not warranted. During the past 4 years, the authors have been devoting study to this phase of relief of overpressure and are now of the belief that even such extreme conditions as fire exposure can be foreseen and provided for. Our studies have covered the effects of exterior heating by small fires and conflagrations, the heat-absorbing capacity of pressure vessels with differing contents, and the capacity necessary in the form of pressure-relief devices; and it is now our desire to present the results for general consideration.

From the observation that pressure vessels surrounded by fire, as in conflagrations, represent what is probably the most hazardous condition to which they may be subjected, the primary objective in the investigation mentioned has been to determine the resulting rise in pressure and the extent of relief capacities necessary to limit pressure rises to safe values. It is believed that, if we thus provide adequately for the worst possible condition, more complete safety can be assured, and that some judicial body, such as the A.S.M.E. Code Committee, will be able to determine what measures are necessary for intermediate cases. It will thus be evident that the primary condition to be studied was that of a pressure vessel filled or partly filled with a volatile liquid and completely surrounded by flame. With this object in view, the foregoing tests (summarized in Part 1) were carried out.

A heat-input rate having been concluded in Part 1, it now becomes necessary to correlate the several variables of container dimensions, working pressures, and properties of contents into useful expressions whereby the necessary relief capacity can be calculated in terms of relief area. This will be done in this section for vessels to operate at pressures above 15 psig.

Summary

Developments herein are based upon the premise that the liquid contents of vessels involved in fire will absorb heat at the rate of 20,000 Btu per hr per sq ft of wetted surface exposed, as developed in Part 1 of this paper.

It was concluded that two formulas, which are derived in the appendix, if properly applied to any vessel to operate at pressures above 15 psig, will specify the relief capacity in terms of area of opening required to limit the pressure rise within 10 per cent above the maximum allowable working pressure. These are as follows:

The diameter of a relief connection (a short tube) necessary to relieve a pressure vessel adequately, when the volatile-liquid contents are vaporized by the heat of the external fire, is determined by the following formula

$$d = \sqrt{\frac{97.3\, S_w}{(P + 14.7)} \left(\frac{P + 448}{3333.3}\right)} \qquad [7]$$

[15] Above 15 psig.

[16] "A.S.M.E. Code for Unfired Pressure Vessels," New York, N. Y., 1940 edition, p. 2.

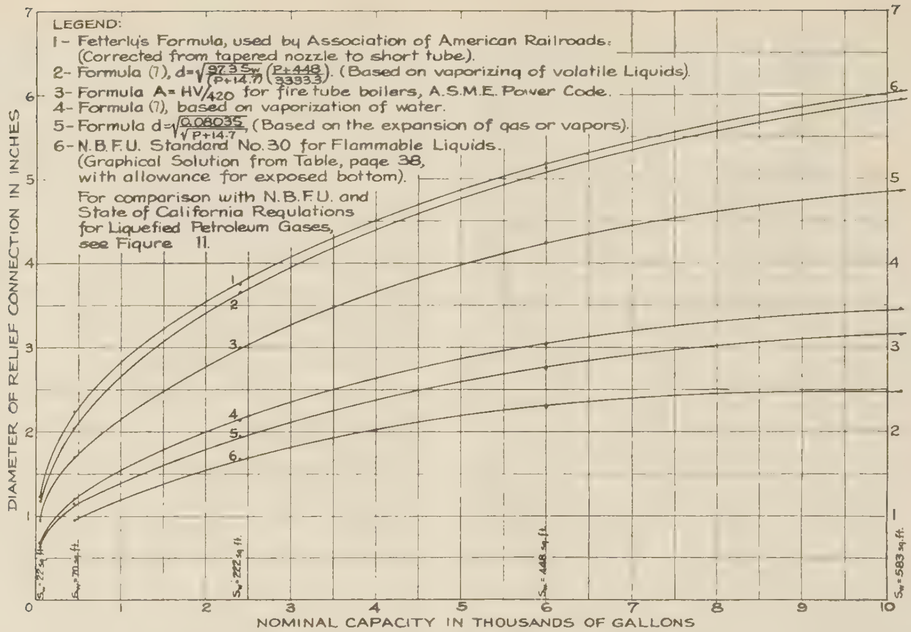


Fig. 10 Comparison of Required Diameter of Connection to Relieve Pressure Vessels of 15 Psig Working Pressure When Exposed to Fire

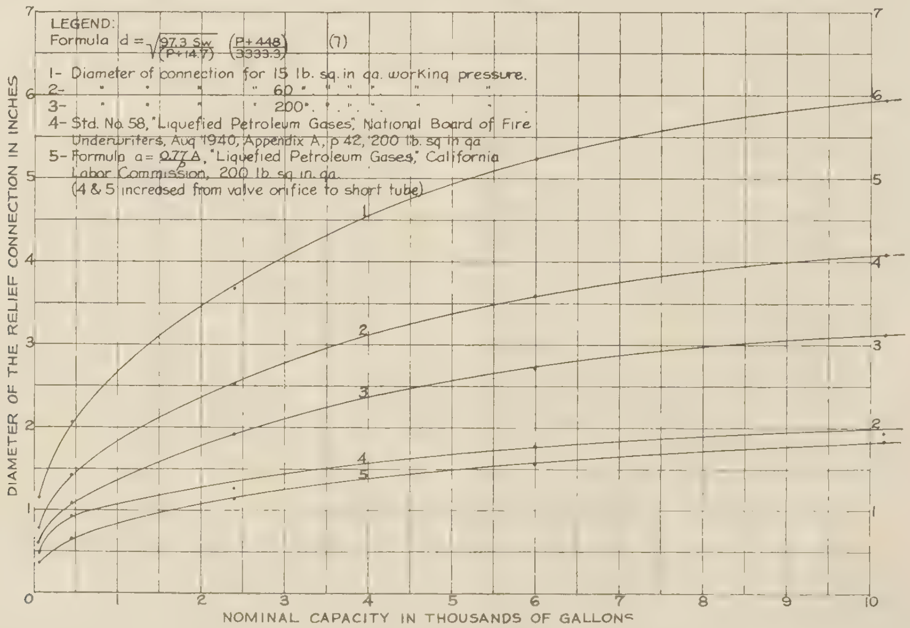


Fig. 11 Comparison of Required Diameter of Relief Connection at Different Working Pressures for Vessels Exposed to Fire

The diameter of a relief connection (a short tube) necessary to relieve a pressure vessel adequately when a gas or vapor (the only contents) is expanded by the heat of external fire is determined by the following formula

$$d = \sqrt{\frac{0.0803\,S}{\sqrt{(P + 14.7)}}} \qquad [8]$$

in which d is in inches, S_w under Equation [7] is wetted surface exposed in square feet, S under Equation [8] is total surface exposed in square feet, and P in either case is maximum allowable working pressure in pounds per square inch gage plus 10 per cent or the safety-device opening pressure, whichever is the lowest. The P in the second term of Equation [7] is not to exceed the numerical value of the critical pressure in pounds per square inch gage of the contents regardless of the working pressure.

In order that the results of these formulas may be compared with other known proposals or standards, Fig. 10 has been in-

cluded. In this it may be noted that the specifications for tank cars of the Association of American Railroads[17] (Fetterly's formula analyzed in Part 1) give relief areas (average over the range) about 10 per cent in excess of Equation [7]; that Equation [7] for volatile liquids gives relief areas about 50 per cent (in larger sizes) greater than those specified by the A.S.M.E. Code[18] for steam; and Equation [7] when modified by the properties of water, gives relief areas about 50 per cent less than the regulations for steam boilers. Although comparison of pressure vessels with boilers is hardly justified (wetted surface exposed to fire compared with boiler heating surface), it does show that the developed formulas give results in the logical direction. (About 6 times as much heat is required to vaporize water as to vaporize an equivalent weight of the volatile liquid considered.)

For comparison of Equation [7] with the National Board of Fire Underwriters' Regulations[19] and the Safety Orders by the Industrial Accident Commission of the State of California[20] for liquefied petroleum gases, see Fig. 11.

Analysis and Method

The reasoning in arriving at Equations [7] and [8] is as follows.

When a vessel is enveloped with flame, heat is absorbed by the contents and temperature within rises. If the contents be liquid, vapors are formed and boiling eventually occurs. The relief capacity must then be at least equal to the vaporization rate, or internal pressure will continue to rise and rupture may follow. If the weight rate of the relief connection is equated to the weight rate vaporized, Equation [9] results which is the origin of Equation [7]

$$d = \sqrt{\frac{IS_w}{205.5P}} \left(\frac{T}{Mr^2}\right)^{1/4} \qquad [9]$$

in which I is the heat-input rate in Btu per hour per square foot, S_w is surface as in Equation [7], P is absolute pressure; and the last term is the contents factor F, in which T is the boiling point at the pressure P in deg F absolute, r is the latent heat of vaporization in Btu per pound at temperature T, and M is the molecular weight. The simplification of this factor is outlined under the heading *Contents Factor*, to be discussed later.

If a vessel contains gas or vapor and is exposed to fire, rapid expansion of the contents takes place (refer to the pressure curve, Fig. 2, Part 1). The relief capacity must then be at least equal to the rate of volume increase, or uncontrolled pressure rise results. If the weight rate of the relief connection is equated to the weight rate of expansion, the following equation results, which is the origin of Equation [8]

$$d = 0.02615 \frac{S^{1/2}}{P^{1/4}} \cdot \frac{(T_w - T)^{5/8}}{T^{0.325}} \qquad [10]$$

in which S is surface as in Equation [8], P is absolute pressure, and the last term is the gas-expansion variable, in which T_w is the temperature to which the tank shell is heated, and T is the initial gas or vapor temperature both in deg F absolute. This term is reduced to a constant by inserting temperature limits, as described under *Gas Expansion Variable*, to be discussed. The rate of volume increase of a gas is always less (other conditions being equal) than the vaporization rate of liquids as will be seen.

For derivations of Equations [9] and [10] refer to the Appendix.

The relief capacity is proposed in terms of area of the relief connection (or connections) because the designer is primarily interested in selecting an adequate fitting of standard size for the vessel to be constructed, and because of the great variation in the capacity of commercial relief apparatus. This connection is ordinarily an inserted short tube with a square inner edge, somewhat rough from the standpoint of fluid flow. The relief device may afterwards be selected of a capacity (based on approved flow test) to pass the vapor rate of flow computed from relief area, as will be explained later. Although the vapors and gases to be contained and discharged have been assumed as perfect gases, the formulas give results in the direction of safety when there is deviation from the gas law.[21]

Weight Rate of a Short Tube. The formula[8] $W = 306\, aP \times \sqrt{M/T}$, which is generally known, is used in this development. It is universally employed and of proved accuracy. This expression is incorporated in the "API-ASME Code for Unfired Pressure Vessels," as the calculated capacity of relief devices of nozzle design. It was derived from the adiabatic flow of an ideal gas through an orifice, modified for flow above the critical-pressure ratio, and holds for this class of equipment having working pressures above 15 psig.

As written, the formula incorporates an orifice coefficient of 0.97 for a tapered nozzle. Generally, 0.83 is used for a short tube, such as the relief connection described (at pressures of this magnitude). Thus the constant has been multiplied by 0.83/0.97 so that the formula reads

$$W = 262\, aP \sqrt{\frac{M}{T}} \qquad [11]$$

for the vapor weight rate of a short tube in pounds per hour. Refer to the Appendix for nomenclature.

The Contents Factor F, $\left(\frac{T}{Mr^2}\right)^{1/4}$ · To use Equation [9], $d = \sqrt{\frac{IS_w}{205.5P}} \left(\frac{T}{Mr^2}\right)^{1/4}$, it is necessary to find values of temperature and latent heat of vaporization for the compounds contained, compatible with the set pressure of the device, P. For many liquids (at various pressures) common to industry, these are not available; and if they were, their use in individual cases would result in numerous vent sizes adequate only for materials of like properties at the same condition. In an effort to facilitate the use of the expression, several studies were undertaken.

Finally, it appeared possible that, if the properties of the known hazardous liquids were graphed, a value of the factor F might be chosen which would be satisfactory for the many compounds to be contained. Accordingly, Fig. 12 was prepared. This is a plot of the numerical value of the contents factor versus the vapor pressure of 36 compounds so far as latent-heat data are available. In so far as possible, a cross section was shown of the many types of liquids and gases commonly found in industry. If the graph were enlarged, additional curves at low pressures could be made intelligible in the more congested area for materials such as hexane, ethylene oxide, styrene, ordinary motor gasoline, isobutylene, vinyl chloride, and "Tetralin." Although of limited number, those shown are sufficient for the development of the discussion.

[17] "Specifications for Tank Cars," Standard, Association of American Railroads, Chicago, Ill., 1941 edition, p. 163.

[18] Minimum Total Areas in Fire Tube Boilers for Safety Valve Connection, "A.S.M.E. Code for Power Boilers," New York, N. Y., 1940 edition, p. 108.

[19] Standards of the National Board of Fire Underwriters, No. 58, "Liquefied Petroleum Gases," New York, N. Y., August, 1940, Appendix A, p. 42.

[20] "Handbook of Butane and Propane Gases," by George H. Finley, editor, Western Business Papers, Inc., Los Angeles, Calif., second edition., 1935, p. 297.

[21] "Discharge Capacity of Relief Valves for Oil Stills," by K. S. M. Davidson and D. W. MacArdle, reprinted from *Oil and Gas Journal*, by Crosby Steam Gauge and Valve Company, Boston, Mass., August, 1929.

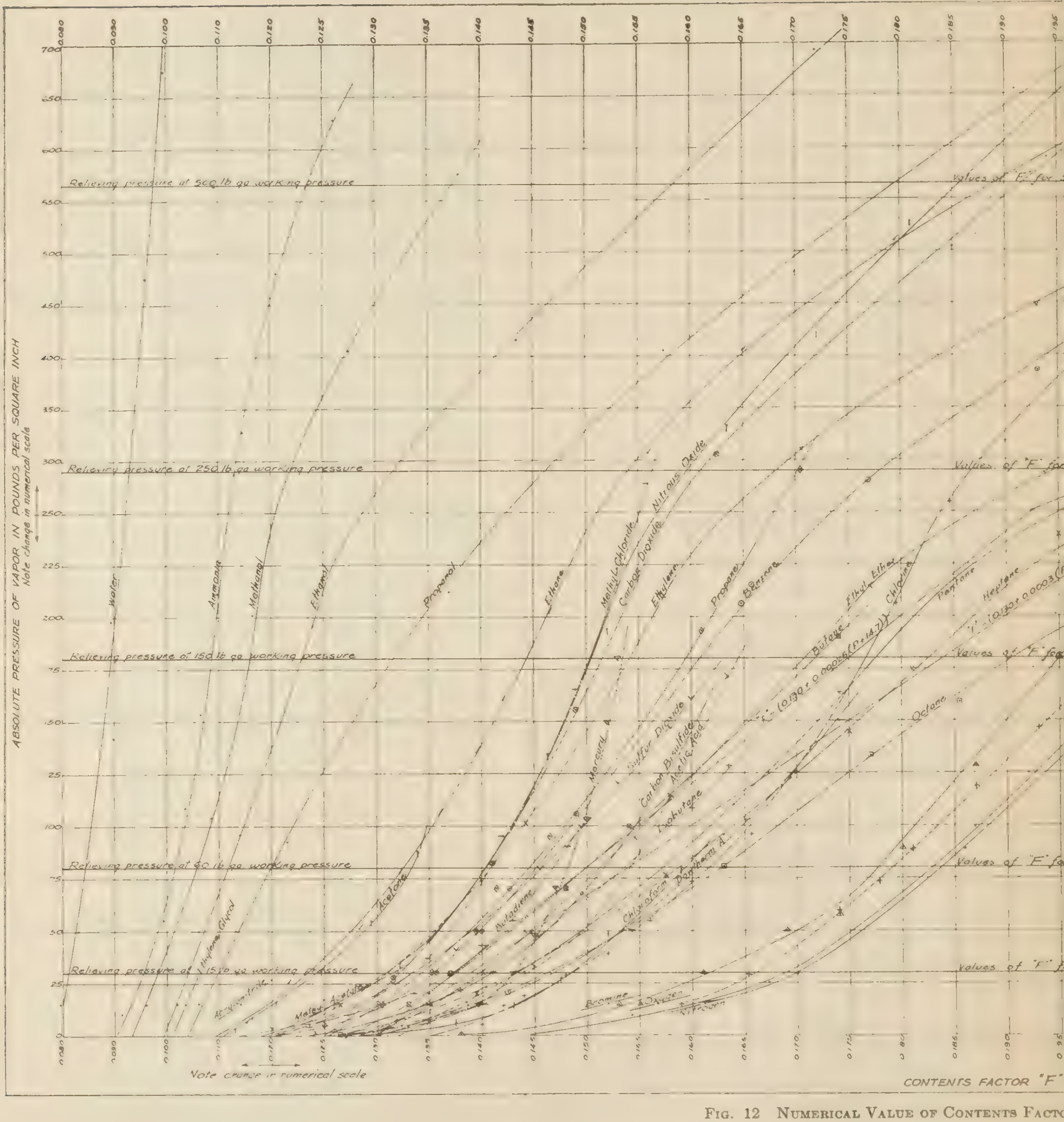


FIG. 12 NUMERICAL VALUE OF CONTENTS FACTO

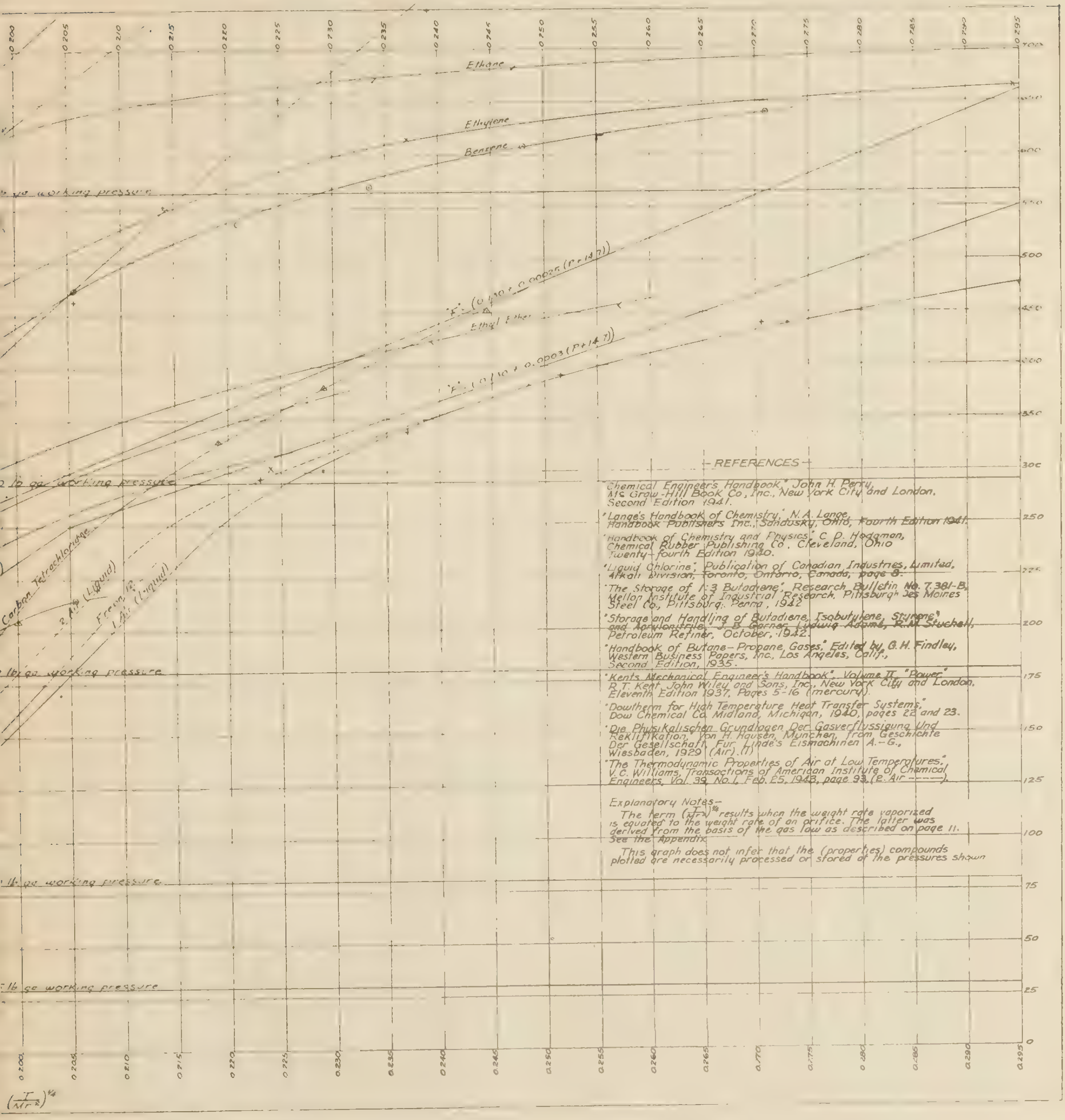
Latent-heat data may be calculated with satisfactory accuracy for other liquids of particular interest by any of several methods found in recent literature. Those provided by H. P. Meissner[22] and Donald F. Othmer[23] are recommended.

Further, the value of F for any compound at 15 psia may be quickly estimated by Trouton's rule, under which conditions F is equal to $(M/T\ 484)^{1/4}$, in which M is the molecular weight and T is the normal boiling point in deg F absolute. It would then be expected from the similarities in the curves in Fig. 12 that the loci of values for the compound considered would approximately parallel those of the compound shown on the graph which is most like that considered in chemical structure and properties.

Where two or more compounds are contained in a vessel, the value of the contents factor should be based on that which will produce the greatest vaporization rate when the volatility and weight per cent of the several components are considered.

[22] "Latent Heats of Vaporization," by H. P. Meissner, *Industrial and Engineering Chemistry*, vol. 33, 1941, p. 1440.

[23] "Correlating Vapor Pressure and Latent Heat Data," by D. F. Othmer, *Industrial and Engineering Chemistry*, vol. 34, 1942, p. 1072.



VARIOUS LIQUIDS AT ABSOLUTE PRESSURES

To illustrate the relative effect of type of contents and magnitude of pressure on the required relief area, Table 2 is given. The diameters were calculated for a large pressure vessel by Equation [9], using values of the contents factor from the graph. The multiplicity of the sizes resulting, the increase of relief diameter with ratio of M/T of the compounds, and the decrease of relief diameter with pressure in the ordinary ranges are then readily apparent. (The decrease in relief diameters above 200 psi working pressure is slight.)

To simplify the use of these data, it has been suggested that similar tables for various sizes of vessels be made, or that the properties of a liquid such as benzene be employed as a standard for values of the contents factor. However, these do not promote what appears to be the more practical manner of assuring adequate relief area for the contingency of fire exposure. The authors suggest that an arbitrary value of F be used which would specify minimum relief-connection size for vessels of any given capacity within certain ranges of pressure. This will also provide an adequate relief connection (within the limits shown in Table 2) for that class of equipment which undergoes changes in service

TABLE 2 ILLUSTRATING EFFECT OF DIFFERENT CONTENTS AND WORKING PRESSURES ON CALCULATED SIZE OF RELIEF CONNECTIONS FOR A 10,000-GAL PRESSURE VESSEL[a]

	Working pressure, psig					Critical temperature, deg F	Critical pressure, psia
	15	60	150	250	500		
Contents	Std. I.P.S., in.			X.S. I.P.S., in.			
Water	3.5	2.5	1.5	1.25	1.0	705.2	3198
Ammonia	4.0	3.0	2.0	1.5	1.25	270.3	1640
Methanol	5.0	3.0	2.0	2.0	1.5	464.0	1155
Ethanol	5.0	3.0	2.5	2.0	1.5	469.6	925
Ethylene glycol	5.0	...	...	...	...	806.0[b]	...
Propanol	5.0	3.5	2.5	2.0	2.0	506.7	733
Ethylene oxide	5.0	...	...	...	...	377.6	...
Acrylonitrile	6.0	...	...	...	...	...	...
Ethane	6.0	4.0	3.0	2.5	2.0	89.8	717
Acetone	6.0	4.0	...	...	...	455.0	690
Ethylene	6.0	4.0	3.0	2.5	2.5	49.5	746
Methyl acetate	6.0	...	...	...	...	452.7	678
Carbon dioxide	6.0	4.0	3.0	2.5	2.0	88.0	1070
Methyl chloride	6.0	4.0	3.0	2.5	...	289.6	965
Sulphur dioxide	6.0	4.0	3.0	...	...	315.0	1140
Propane	6.0	4.0	3.0	2.5	...	206.2	617
Nitrous oxide	6.0	4.0	3.0	2.5	...	97.7	1052
Benzene	6.0	4.0	3.0	3.0	2.5	551.3	701
Isobutylene	6.0	...	...	...	...	...	...
Butadiene	6.0	4.0	...	...	...	325.7	...
Butane	6.0	4.0	3.0	...	...	307.4	529
Styrene	6.0	...	...	...	...	671.0[b]	...
Ethyl ether	6.0	4.0	3.0	3.0	...	380.8	521
"Tetralin"	6.0	...	...	...	...	833.0[b]	...
Isobutane	6.0	4.0	...	...	...	273.2	544
Carbon bisulphide	6.0	4.0	...	...	...	523.4	1116
Mercury	6.0	4.0	3.0	...	...	>2822.0	>2940
Heptane	6.0	4.0	3.5	3.0	...	512.2	394
Pentane	6.0	4.0	3.5	3.0	...	387.0	485
Hexane	6.0	...	...	...	...	454.6	434
Acetic acid	6.0	4.0	3.0	...	...	610.9	839
Octane	6.0 and 2.0	5.0	3.5	...	...	564.8	361.5
"Dowtherm A"	6 0 and 2.0	5.0	...	...	...	968.0	...
Chloroform	6.0 and 2.0	5.0	...	...	...	505.4	...
Gasoline (ordinary)	6.0 and 2.0	...	...	...	...	...	...
Chlorine	6.0 and 2.0	5.0	3.5	3.0	2.5	291.2	1118
Bromine	6.0 and 3.5	...	...	...	...	576.0	...
Carbon tetrachloride	6.0 and 3.5	5.0	3.5	3.0	...	541.6	660
"Freon 12"	6.0 and 4.0	5.0	3.5	...	...	...	...
Air (liquid)	6.0 and 4.0	5.0	3.5	3.5	3.5	—253.2	547
When $F = 0.130 + 0.0003 (P + 14.7)$	6.0	4.0	3.5	3.0	3.0	...	...
When $F = 0.130 + 0.00025 (P + 14.7)$	6.0	4.0	3.0	3.0	3.0	...	...

[a] Dimensions of pressure vessel, 10 ft 6 in., diam; 18 ft 10 in., height.

[b] Approximate.

NOTE: This table does not infer that the compounds shown are necessarily processed or stored at the pressures listed. Calculations are based on the set pressure of the relief device being 10 per cent in excess of the working pressure.

and which would be desirable from the standpoint of maintenance and operation.

Values of F which might be used to specify the relief-connection size are represented by the designated straight lines in Fig. 12. Reference to Table 2 will show that practical values of d are obtained when F equals $0.130 + 0.00025\ (P + 14.7)$ or $0.130 + 0.0003\ (P + 14.7)$, and when relief devices are applied as suggested in the section, Summary for Use. For purposes of facilitating this development, it is the latter value of F which has been employed in simplifying Equation [9] to Equation [7], since it not only appears adequate, as will be explained, but is not exceeded (except by liquid air),[24] within the limits of available data at higher pressures by the individual contents-factor curves. Very probably, the relief area for pressure vessels larger than 10,000 gal nominal capacity should be computed by Equation [9] for the individual contents when the necessary latent heat-vaporpressure data are obtainable.

It is to be noted that the suggested value $[0.130 + 0.0003\ (P + 14.7)]$ of F gives diameters consistent with the large majority of the compounds, but less than for some of those tabulated which have high ratios of M/T. Calculations show that this inadequacy holds only at pressures below 100 psig when the next larger pipe size is used as the relief connection. Inasmuch as pressure vessels usually incorporate a strength safety factor of 5, the relief diameters are based upon a pressure increase limited to 10 per cent of the working pressure, and the vessels are assumed to be 90 per cent full of volatile material when exposed to fire, no doubt the straight-line value of F would prove adequate at any pressure. Such a value would also afford the benefits of standardization. It is understood, however, that such standardization will not be wholly agreeable to all concerned, and further improvement in regard to simplifying the contents factor appears warranted. Nevertheless, the suggested value of F in no degree nullifies the value of Equation [9] in calculating the required relief diameter directly for any contents.

Further, it should be noted that the individual curves of the contents factor flatten as the critical pressures of the compounds are approached. It is in this region that the latent heat of vaporization of the liquid approaches zero; and consequently, the individual value of F increases more rapidly so that the relief diameter increases slowly after having decreased rapidly with pressure below 200 lb working pressure. Such increase is in conformance with the trend of accelerated vaporization. However, the value of F should not be increased beyond the point of critical temperature and pressure in those exceptional cases where processes operate at pressures such that the relieving pressure exceeds the critical pressure of the material being processed. Critical pressures and temperatures of representative compounds are listed in Table 2. (Processes which operate altogether above the critical pressure and temperature are in gas service and Equation [10] applies.)

The Gas Expansion Variable, $\dfrac{(T_w - T)^{5/8}}{T^{0.325}}$.

The rate of volume increase just discussed is dependent upon

[24] This is as might be expected; since in the evaluation of the contents factor for gases, $(T/MC_p)^{1/2}$, the numerical value of the factor for air was found to be larger than that for any other gas whose properties were available in the reference literature (refer to the Appendix).

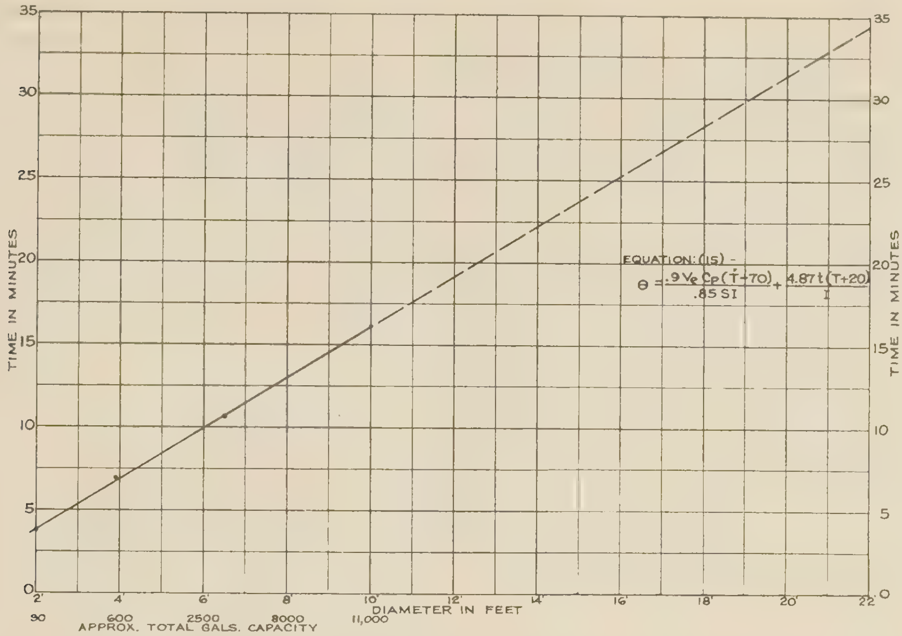


FIG. 13 ESTIMATED TIME FOR VOLATILE LIQUID CONTENTS OF PRESSURE VESSELS TO REACH 158 F WHEN ABSORBING HEAT AT 20,000 BTU PER SQ FT PER HR

(Contents of properties of F equals 0.142, T = 158 F; M = 74; r = 143 Btu; C_p = 0.447; ρ = 54.7 lb, at 15 psig.)

the rate of temperature rise in the contents (gas or vapor). This reduces mathematically (see derivation in Appendix) to the term shown. The probable maximum temperature difference is considered to be 1100 F; that is, $(T_w - T) = (1560 - 460)$ F abs, initial vapor temperature being zero F, and tank-wall temperature rising to 1100 F. Conceivably, this temperature difference may be greater, but distortion and possible failure of steel plate above 1100 F may be expected, even though the internal pressure is limited to a 10 per cent rise. When these temperature limits, typifying severest conditions, are inserted, the variable reduces to the constant 10.85.

The variable was derived using the properties of air in order to obtain maximum values and so provide adequate relief for any gas or vapor. Refer to the Appendix for comparison of values obtained with properties of other gases.

Flow Capacity in Terms of Area

If the relief area (not the next larger pipe size) as just determined, the molecular weight of the contents, and the absolute vapor temperature at the relieving pressure are inserted in weight rate Equation [11] of a short tube (a relief connection), the necessary capacity of the relief device may be computed. The apparatus may then be selected of a capacity (based on approved flow test) to pass the calculated rate which is as follows:

$$W = 262\ aP\sqrt{\frac{M}{T}}\ \text{lb per hr of vapor} \ldots\ldots [11]$$

$$Q = \frac{1682\ aP}{\sqrt{MT}}\ \text{cfm at 68 F of free vapor} \ldots\ldots [12]$$

$$Q^* = \frac{353.0\ aP}{\sqrt{T}}\ \text{cfm of free air at 68 F} \ldots\ldots [13]$$

$$W = 42.7\ aP\ \text{lb per hr of saturated steam} \ldots [14]$$

In all these cases, a is in square inches, P is the opening pressure of the relief device in pounds per square inch absolute, M is the molecular weight of the contents, and T is the contents vapor temperature in deg F absolute at the pressure P.

For conditions of fire exposure, T is approximately the temperature of the liquid contents when P is the vapor pressure, which may be obtained from temperature-vaporpressure data. For vessels in gas or vapor service only, T is approximately the temperature necessary to raise the pressure in the vessel from the absolute working pressure and temperature to the absolute set pressure of the relief device, as computed by the gas law, $T/P = T'/P'$.

The source of Equations [11], [12], [13], and [14] will be found in the Appendix.

It is in this determination of the capacity of the relief device (or devices) that allowance is made for the individual contents and immediate service, it being proposed, as already inferred, that the relief connection be the same size on all vessels of given capacity and range of working pressure. When equipment is so provided with adequate relief area for volatile compounds in the form of relief connections, the relief devices may be replaced to provide for any change in contents or more severe working conditions from the standpoint of relief requirements.

The procedure in determining the size of connections, capacity of devices, and selection of relief apparatus will be outlined in the Summary for Use.

The Time Element

If a vessel exposed to fire is large and the contents are of such properties that there would be a long delay before boiling and maximum efflux would occur, it would be desirable to select the relief device to provide for the vaporization rate which would take place within a certain safe period. The time required for the liquid contents of a steel vessel to reach a given average temperature during fire exposure may be approximated by use of the following formula

$$\theta = \frac{V\rho C_p(T - 70)}{S_w I} + \frac{4.87t(T + 20)}{I} \ldots\ldots [15]$$

in which θ is the time in hours, V is volume of liquid contents in cubic feet, ρ is the density of the liquid in pounds per cubic foot, C_p is the specific heat of the liquid, T is the boiling point at the opening pressure of the relief device or temperature reached in a period θ in deg F, S_w is the wetted surface exposed to fire in square feet, I is the heat-input rate (20,000 Btu per hr per sq ft), and t is the thickness of the tank shell (steel) in inches. Derivation of this expression will be found in the Appendix.

The time required for volatile contents to reach the boiling point during fire exposure is very short. To illustrate, Fig. 13 has been included. It was assumed that the vessels were 90 per cent full with 85 per cent of the total surface wetted and exposed—a quite normal condition. Properties of the contents were taken as those having an F value of 0.142, and as otherwise noted in the caption of Fig. 13. Under these conditions, a short time element and a maximum venting rate may be expected. It will be seen that the time element is very largely a function of the diameter of the vessel. The calculations indicate that most (those under 20 ft diam) pressure vessels with volatile contents at 15 psi will boil when exposed to intense fire in less than $^1/_2$ hr.

When vessels are in gas or vapor service and become exposed to fire, the maximum venting rate will be reached within such a short period as to permit no reduction of the relief capacity as calculated by Equation [8]. Fig. 2, Part 1, shows this very clearly.

Effect of "Outage"

The term "outage" is used in the literature pertaining to containers to indicate the proportion of a tank which contains vapor rather than liquid. Thus, a tank which is only 40 per cent filled would have an "outage" of 60 per cent. Although this term does not appear in the developed formula, it has been considered. If the wetted height of the tank shell is used rather than the nominally full wetted height, the effect of "outage" is provided for. It is true that as the "outage" increases, the unwetted wall temperature will rise; but heat will be lost by conduction and radiation to the colder portions of the tank, and the hot wall temperature (steel) will probably not rise above the "critical" point of about 1100 F. When the "outage" is large, the wetted height is small, and the rate of vaporization is low; and although the vapor temperature will be higher than the temperature corresponding to the boiling point, it can be shown that the highest venting rate will occur when a tank is full of liquid.

The time required for a completely empty steel tank to reach 1100 F in a fire of intensity I would be approximately

$$\theta = \frac{5144\, t}{0.8 I} \text{ or } 0.320\, t \text{ for 20,000 Btu input rate} \quad [16]$$

in which θ is time in hours for the tank wall to reach 1100 F, t is thickness of tank in inches, and I is intensity of fire in Btu per hr per sq ft. Thus in a fire of the intensity of 20,000 Btu per hr per sq ft, a tank made of $^1/_4$-in. steel would reach 1100 F in about 5 min.

$$\theta = 0.320 \times 0.25 = 0.080 \text{ hr} = 4.8 \text{ minutes}$$

This agrees very closely with time-temperature observations made by the Underwriters' Laboratories, Inc., in their tests previously referred to.[10] The source of this formula will be found in the Appendix.

Effect of Ranges of Pressure on Relief Areas

As mentioned previously, the internal diameters of the required relief connections vary rapidly with changes in working pressure up to about 200 psig. To show this clearly, Equation [7]

$$d = \sqrt{\frac{97.3\, S_w}{(P + 14.7)}\left(\frac{P + 448.0}{3333.3}\right)}$$

for vessels in volatile-liquid service has been graphed at 15, 60, and 200 psig working pressure, Fig. 11. Above 200 psi, a very gradual change in the required relief diameter continues, but the pipe size will not be affected for vessels of 10,000 gal nominal capacity or less until pressures above 700 psig are reached.

It is seen from Fig. 11 that it would be practical and economical to use different size relief connections for different ranges of working pressure on vessels of the same capacities. Further examination of the graph will disclose that the curves at the pressures indicated conform largely (more so than at other pressures when trial calculations are made) to pipe sizes. To take advantage of this effect, relief-connection formulas for ranges of working pressure will be provided in the Summary for Use.

In order to complete the comparison described under the Summary, of Part 2, the relief sizes of the National Board of Fire Underwriters and the State of California for liquefied petroleum gases at 200-psig working pressure have been added to Fig. 11. The difference between these and Equation [7] is principally due to the higher heat-absorption rate, although a somewhat larger contents factor is used in Equation [7].

A study of the shapes and proportions of vessels will disclose, as mentioned previously, that when calculating relief areas for connections, it is always adequate to use 85 per cent of the total exterior surface as the wetted area, provided the normal filling is not more than 90 per cent of the total volume. This has been employed in providing formulas for sizes of connections.

Emergency and Normal Relief

It is to be recognized that the requirement for relief of overpressure due to fire exposure represents an abnormal condition, and that this requirement, in most cases, is greatly in excess of that required to provide for operation fluctuations or disturbances. This latter requirement is considered by the authors to be "normal" relief and the former "emergency" relief. More specifically, normal relief is that required by process overheating, overfilling, improper operation, failure of control apparatus, the human element, plugging or mechanical failure within the system, etc. Emergency or abnormal relief is considered that due to fire exposure or uncontrolled, internal, chemical reaction.

The two present separate problems. Required normal relief can be determined from analysis of physical conditions within the system and calculation of the possible maximum "through-put." Relief required, due to fire exposure, may be provided for as outlined in this paper. As stated, the two requirements are widely separated in most cases, and it is practical to provide separate devices for the purposes as will be proposed.

Internal reaction is an emergency requirement not easily determined. The multiplicity and range of chemical reactions or rapid decompositions defy classification of rates of volume increase. Where likelihood of such expansion exists, special analysis and provisions are necessary to safeguard the individual installation.

On the other hand, there are special installations and processes in which the normal relief is the larger requirement. Some of these may be (depending upon analysis) vessels covered with substantial, fire-resistive insulation, those partly so insulated, vessels protected with reliable cooling systems, vessels processing nonhazardous plastics and semi-solids, etc. In some such cases, it will be apparent after analysis that no supplemental relief capacity need be provided for fire exposure as long as the immediate service is not made more severe from the standpoint of fire contingency; but these allowances in relief-device capacity should not effect a reduction in the area of the relief connection for fire exposure as previously specified. When vessels are partly protected with nonflammable insulation, the formulas may be correctly applied to the exposed surface in determining the necessary capacity of the relief device.

Relief due to fire exposure is assumed to be the release of vapor only. Liquid expansion due to the increased temperature is relatively negligible.

Proposed Summary for Use

1 Minimum sizes of relief connections for vessels not exceeding 10,000 gal nominal capacity (585 sq ft of wetted surface exposed): All pressure vessels should be provided with an emergency relief area (or areas) in the form of nozzles or connections. This area should be that computed by the following formulas or that required to limit the rise in working pressure to 10 per cent (as based on maximum possible "through-put" due to operation fluctuations or disturbances as defined), whichever of the two is the larger:

For vessels to have a working pressure from 15 to 60 psig, use

$$a = 0.0405\,S \text{ or } d = \sqrt{0.0515\,S} \quad \text{[1]}'$$

For vessels to have a working pressure from 60 to 200 psig, use

$$a = 0.0192\,S \text{ or } d = \sqrt{0.0244\,S} \quad \text{[2]}'$$

For vessels to have a working pressure from 200 to 700 psig use

$$a = 0.01112\,S \text{ or } d = \sqrt{0.01415\,S} \quad \text{[3]}'$$

In Equations [1]′, [2]′, and [3]′, a is the relief area (or total areas) in square inches, S is the total external surface of the vessel in square feet, and d is the internal diameter of the circular relief opening in inches.

For vessels to have a working pressure above 700 psig, use

$$d = \sqrt{\frac{97.3\,S_w}{(P + 14.7)}\left(\frac{P + 448}{3333.3}\right)} \quad \text{[4]}'$$

in which d is as just given, P is the set pressure of the relief device (normally 1.1 the working pressure) in pounds per square inch gage, and S_w is the surface in square feet wetted by the contents and exposed to fire, P in the second term is limited to the critical pressure of the compound in process.

For vessels larger than 10,000 gal nominal capacity (585 sq ft of wetted surface exposed), use

$$d = \sqrt{\frac{97.3\,S_w}{(P + 14.7)}\left(\frac{T}{Mr^2}\right)^{1/4}} \quad \text{[5]}'$$

in which the symbols are as in the preceding paragraph except those in the last term which is the contents factor. It should be evaluated directly from Fig. 12, or calculated as recommended under the heading *Contents Factor*. In all cases, the nearest suitable pipe size (or sizes) of adequate area should be used.

2 Relief capacity in terms of area required at any pressure; liquid service: All pressure vessels in liquid service, or if used in systems which contain liquids, should have a relief flow capacity equivalent to the internal cross-sectional area (not the nearest pipe size) of a short tube, as calculated by the following formula, or that area required for "normal" relief, as described in paragraph 1, whichever is the larger

$$d = \sqrt{\frac{97.3\,S_w}{(P + 14.7)}\left(\frac{T}{Mr^2}\right)^{1/4}} \text{ and } a = 0.7854\,d^2 \quad \text{[6]}'$$

In Equation [6]′, a is the relief area (or total of areas) in square inches, S_w is the wetted (wetted by the liquid contents when nominally full) surface exposed in square feet, d is the inside diameter of the free circular opening in inches, and P is the maximum allowable working pressure plus 10 per cent in pounds per square inch gage. The term $\left(\frac{T}{Mr^2}\right)^{1/4}$ is the contents factor and is evaluated as described in the preceding paragraph under Equation [5]′. (This is the emergency relief capacity required for vaporization of liquids due to fire exposure.)

Exceptions: In those exceptional cases where a, as determined by Equation [6]′, exceeds the area of the relief connection, as set forth in paragraph 1, the relief connection will have to be increased or allowance permitted as suggested near the end of the section *The Contents Factors F*, $\left(\frac{T}{Mr^2}\right)^{1/4}$. If the relief capacity in terms of area approaches that of the relief connection, multiple relief devices will be required and should be applied as described in paragraph 5.

3 Relief capacity in terms of area required at any pressure; gas or vapor service only: All pressure vessels in gas or vapor service and in gas systems where no liquid can accumulate should have a relief flow capacity equivalent to the internal cross-sectional area of a short tube, as determined by the following formula, or that area required for "normal" relief, as described in paragraph 1, whichever is the larger

$$a = \frac{0.0631\,S}{\sqrt{(P + 14.7)}} \quad \text{or} \quad d = \sqrt{\frac{0.0803\,S}{\sqrt{(P + 14.7)}}} \quad \text{[7]}'$$

In Equations [7]′, a is the relief area in square inches, S is the external exposed surface in square feet, d is the inside diameter of the relief connection in inches, and P is the maximum allowable working pressure plus 10 per cent in lb per sq in. gage. This is the emergency relief capacity required for gas or vapor expansion due to fire exposure.

4 Relief capacity of the required areas in terms of fluid flow: To select a relief device (or devices) of adequate capacity (based on approved flow test), use the area a, as determined in paragraph 2 or 3, in any one of the following formulas as the case may require:

$$W = 262.0\,aP\sqrt{M/T} \text{ lb of contents vapor per hr} \quad \text{[8]}'$$

$$Q = \frac{1682\,aP}{\sqrt{MT}} \text{ cfm of vapor at 68 F and atmospheric pressure} \quad \text{[9]}'$$

$$Q = \frac{353.0\,aP}{\sqrt{T}} \text{ cfm of air at 68 F and atmospheric pressure} \quad \text{[10]}'$$

$$W = 42.7aP \text{ lb per hr of saturated steam} \quad \text{[11]}'$$

In Equations [8]′, [9]′, [10]′, and [11]′, a is in square inches, and P is the opening pressure of the relief device in pounds per square inch absolute; M is the molecular weight of the contents, and T is the contents vapor temperature in deg F absolute at the pressure P. For conditions of fire exposure, T is approximately equal to the temperature of the liquid contents when P is the vapor pressure, which temperature may be obtained from temperature-vapor pressure data. For vessels in gas or vapor service only, T is approximately the temperature necessary to raise the pressure in the vessel from the absolute working pressure and temperature to the absolute set pressure of the relief device, as computed by the gas law, $T/P = T'/P'$.

5 Normal and emergency relief devices: It is preferable that separate spring- or weight-loaded safety devices be used for the "normal" relief requirement as described under that heading. The additional devices necessary to provide for fire exposure or internal reaction may supplement those required for operation fluctuations or disturbances. These may be in the form of rupture disks, safety heads, pilot-operated valves, or other apparatus and should be installed as specified in the recognized codes. If

rupturing devices are used, they should be calibrated to release at a pressure 50 to 100 per cent above the working pressure but not to exceed the test pressure of the vessel. In the exceptional cases where the "normal" and "emergency" relief are so nearly equivalent as to make the installation of multiple devices not economical, the one device should be of the capacity of the larger of the two requirements and set to relieve 10 per cent above the working pressure.

In those cases where the relief for operation fluctuations may be the larger requirement because of substantial fire-resistive insulation, nonhazardous contents or surroundings, etc., Equations [6]′ and [7]′ may be applied on the basis of the part of the surface not covered, or the part possible to be exposed when arriving at the relief capacity.

6 Back pressure: The pressure on the downstream side of the relief device should be limited to a maximum of from 10 to 25 per cent of the set pressure which must be adjusted to counteract the back pressure. If this limit is exceeded the capacity of the relief device may be reduced and redesign become necessary.[21]

7 Lower limit of size: For reasons of mechanical strength, it is recommended that no relief connection or safety device for manufacturing or service equipment be smaller than 1/2 in. pipe size, regardless of the size of vessel to be protected.

8 Time element: To estimate the time for volatile liquids in a vessel exposed to fire to reach a given average temperature, apply the following formula

$$\theta = \frac{V\rho C_p(T - 70)}{S_w\, 20{,}000} + \frac{4.87\, t(T + 20)}{20{,}000} \quad \ldots\ldots [12]'$$

In Equation [12]′, θ is the time in hours, V is the volume of the liquid in cubic feet, ρ is the density of the liquid in pounds per cubic foot, C_p is the specific heat of the liquid, T is the boiling point at the opening pressure of the relief device, or temperature reached in a period θ in deg F, S_w is the wetted surface exposed in square feet and t is the thickness of the steel shell in inches.

If θ is 1/2 hr or more (approximate time element for 20-ft-diam vessels containing volatile liquids at just above 15-psig working pressure) and there is assurance that effective cooling or fire-extinguishing equipment will come into operation during the half-hour period, the emergency-relief capacity may be reduced to the vaporization rate produced by the temperature rise within the half hour. For vessels exposed to fire in gas service, the maximum venting rate will occur within such a short time as to permit no modification of the calculated emergency relief.

Conclusion

The conclusions as outlined under Summary and detailed under Summary for Use are based upon the foregoing theory which is mathematically expressed in the Appendix.

The formulas proposed give relief capacities much greater than are at present employed and which are not adequate for the abnormal condition of fire exposure.

For example of use of the formulas refer to the Appendix.

Proposals herein hold only for pressures above 15 psig. For vessels operating at pressures not exceeding 0.50 psig, Part 3 will give details. Part 4 will appear at a later date covering vessels operating from 0.50 through 15 psig.

In addition to the lack of complete specifications in the recognized codes for the pressure-relief requirements of vessels, and the general statement that present relief areas as proposed by regulatory bodies were found to be inadequate for fire exposure, it has been suggested that further information be given to justify this study which shows a method of computing relief capacities for the contingency of fire exposure.

In its present form, the paper is based upon the mathematical development of the observations of vessels exposed to seven man-made fires by four separate organizations, as described in Part 1 of this paper. Since its preparation, two other similar tests by another company have been received, which support the tests herein analyzed.

Also in evidence are the reports of four equipment failures from accidental fire. These were vessels constructed in accordance with the A.S.M.E. Unfired Code and provided with relief areas compatible with a heat-absorption rate when exposed to fire of 7000 Btu per sq ft of wetted surface per hr. Three contained stable compounds, and the contents of the fourth probably underwent partial dissociation above the boiling point. All of the several contents had low-vapor pressures, boiling at much higher temperatures than water. No exact data on the fire intensity were obtained, but trial calculations by the developed formulas, using what data were available, gave results favorable to the test observations.

Further, the authors have reports of 31 failures of equipment (not due to fire exposure) other than those which may be obtained from several recording associations. Of these, 20 were due to excessive external pressure and 10 from excessive internal pressure. The overpressures occurred from miscellaneous causes for which the "normal" relief capacity should have been adequate, and it is not doubted that many of the failures would have been prevented had relief capacity for fire exposure also been incorporated in the installations.

Since it is not possible to specify a "normal" relief requirement for pressure vessels except by individual analysis of physical conditions within the process, and since it is possible to determine and provide relief capacity for fire exposure in a practical and relatively inexpensive manner, it is felt that this paper may provide a beginning for discussion, compilation of experience, and study which might eventually lead to more specific and safe specifications for minimum pressure-relief requirements.

Appendix to Part 2

Derivation of General Equations [9] or [3]″, and [10] or [15]″

Vapor weight rate of a short tube (above 15 psig)

$$W = 306\, aP \sqrt{\frac{M}{T}}$$

API-ASME Code: Weight rate of tapered nozzle

$$W = 306 \times \frac{0.83}{0.97}\, aP \sqrt{\frac{M}{T}}$$

$$W = 262\, aP \sqrt{\frac{M}{T}}$$

or

$$W = 262 \times 0.7854\, d^2P \sqrt{\frac{M}{T}}$$

$$W = 205.5\, d^2P \sqrt{\frac{M}{T}} \quad \ldots\ldots [1]''$$

in which

(Coefficient of flow of a short tube = 0.83)

W = rate, lb per hr
a = orifice area, sq in.
P = inlet pressure, psia
M = molecular weight of vapor
T = vapor temperature, deg F abs
d = orifice diameter, in.

Weight rate vaporized

$$\frac{Q}{S_w} = \frac{Wr}{S_w} = I$$

$$W = \frac{IS_w}{r} \quad \text{[2]}''$$

in which

- Q = total heat input, Btu per hr
- S_w = wetted surface heated, sq ft
- r = latent heat of vaporization, Btu per lb
- I = unit heat input, Btu per hr per sq ft
- W = liquid vaporized, lb per hr

Required diameter of a short tube to pass the weight rate vaporized: Equate [1]″ and [2]″

$$\frac{IS_w}{r} = 205.5\, d^2P \sqrt{\frac{M}{T}}$$

$$d^2 = \frac{IS_w}{r}\left(\frac{T}{M}\right)^{1/2}\left(\frac{1}{205.5\,P}\right)$$

$$d = \sqrt{\frac{IS_w}{205.5(P + 14.7)}}\left(\frac{T}{Mr^2}\right)^{1/4} \quad \text{[3]}'' \text{ or [9]}$$

To apply Equation [3]″

- d = diameter of relief connection, in.
- I = 20,000 Btu per hr per sq ft
- S_w = wetted surface heated, sq ft
- P = set pressure of device, 1.1 working pressure, psig
- T = boiling point at P, deg F abs
- M = molecular weight of contents
- r = latent heat of vaporization at T

Weight rate of expansion of gas or vapor due to heat transferred to a tank:

$$wC_p\frac{dT}{d\theta} = \frac{Q}{\theta} \quad \text{[4]}''$$

$$hS\,\Delta T = \frac{Q}{\theta} \quad \text{[5]}''$$

$$w = V\rho = V\frac{MP}{RT} \quad \text{[6]}''$$

From Equations [4]″ and [6]″

$$V\rho C_p\frac{dT}{d\theta} = \frac{Q}{\theta} \quad \text{[7]}''$$

$$h = F\overline{\Delta T}^{1/4} \text{ (ref. 5, Equation [16], p. 240)}$$

From Equation [5]″

$$F\overline{\Delta T}^{1/4}S\,\Delta T = \frac{Q}{\theta} \quad \text{[8]}''$$

Equate [7]″ to [8]″

$$V\rho C_p\frac{dT}{d\theta} = FS\overline{\Delta T}^{5/4}$$

$$\frac{dT}{d\theta} = \frac{FS\overline{\Delta T}^{5/4}}{V\rho C_p} \quad \text{[9]}''$$

From Equation [6]″

$$dw = \frac{-VMP}{R}\cdot\frac{dT}{T^2} \quad \text{[10]}''$$

$$-dw = \frac{w}{\theta}\,d\theta \quad \text{[11]}''$$

Equate [10]″ to [11]″

$$\frac{VMP}{R}\cdot\frac{dT}{T^2} = \frac{w}{\theta}\,d\theta \quad \text{[12]}''$$

$$\frac{dT}{d\theta} = \frac{w}{\theta}\cdot\frac{RT^2}{VMP} \quad \text{[13]}''$$

Equate [9]″ to [13]″

$$\frac{FS\overline{\Delta T}^{5/4}}{V\rho C_p} = \frac{w}{\theta}\;\frac{RT^2}{VMP} \quad \text{[14]}''$$

in which

- w = weight of contained gas or vapor, lb
- C_p = specific heat of gas or vapor
- dT = change in temperature
- $d\theta$ = change in time
- Q = total heat transferred, Btu
- θ = time, unit hr
- h = heat-transfer coefficient, Btu per hr per sq ft per deg F
- S = tank surface heated, sq ft
- ΔT = difference between tank wall and contained gas, deg F
- V = volume of contents gas or vapor, cu ft
- ρ = density of contents gas, lb per cu ft
- M = molecular weight of gas or vapor
- P = pressure, psia
- R = gas constant, 10.72
- T = temperature of gas or vapor, deg F abs
- F = function of physical properties of a gas
- dw = change in weight
- $W = \frac{w}{\theta}$ = venting rate, lb per hr

Required diameter of a short tube to pass the weight rate of gas expanded due to heat transferred to a tank (the same nomenclature is used as in the preceding group):

Insert Equation [1]″ in [14]″

$$205.5\,d^2P\sqrt{\frac{M}{T}}\cdot\frac{RT^2}{VMP} = \frac{FS\overline{\Delta T}^{5/4}}{V\rho C_p}$$

$$d^2 = \frac{FS\overline{\Delta T}^{5/4}VMP}{V\rho C_p\,RT^2\,205.5\,P}\left(\frac{T}{M}\right)^{1/2}$$

$$d^2 = \left(\frac{F}{C_pM^{1/2}}\right)\frac{S}{205.5}\cdot\frac{M\overline{\Delta T}^{5/4}}{RT^{3/2}\rho}$$

in which $\left(\rho = \frac{MP}{RT}\right)$

$$d = \left(\frac{F}{C_pM^{1/2}}\right)^{1/2}\cdot\frac{S^{1/2}}{14.33\,P^{1/2}}\cdot\frac{\Delta T^{5/8}}{T^{1/4}}$$

$$d = \left(0.375\,\frac{P^{1/4}}{T^{0.0753}}\right)\cdot\frac{S^{1/2}}{14.33\,P^{1/2}}\cdot\frac{\Delta T^{5/8}}{T^{1/4}}$$

Refer to following paragraph for $0.375\,\frac{P^{1/4}}{T^{0.0753}}$

$$d = 0.02615\,\frac{S^{1/2}}{P^{1/4}}\cdot\frac{\Delta T^{5/8}}{T^{0.325}} \quad \text{[15]}'' \text{ or [10]}$$

To apply Equation [15]″

- d = diameter of relief connection, in.
- S = surface exposed, sq ft

P = set pressure of relief, 1.1 gage working pressure plus 14.7
ΔT = $(T_w - T)$
T_w = temperature to which tank wall is heated, deg F abs
T = initial temperature of tank vapor, deg F abs

The term $\left(\frac{F}{C_pM^{1/2}}\right)^{1/2}$ in the foregoing, relating the properties of the contents fluid, was replaced by its maximum value $0.375\,\frac{P^{1/4}}{T^{0.0753}}$. This is justified in the following manner: The value of F, a function of the physical properties of a gas, varies with both temperature and pressure as will be seen. The value C_p, specific heat at constant pressure, varies with temperature. Inasmuch as pressure and temperature are working conditions to remain in the relief diameter Equation [15]″, it is desirable to express the subject variable in terms of P and T. Now

$$F = D\left(\frac{\rho^2BC_pk^3}{Z}\right)^{1/4} \quad \text{(ref. 5, Equation [20], p. 242)}$$

$$\left(\frac{F}{C_pM^{1/2}}\right)^{1/2} = D\left(\frac{\rho^2BC_pk^3}{C_p^4M^2Z}\right)^{1/8}$$

$$= D\left[\rho^2Z^2\cdot\frac{B}{M^2}\cdot\left(\frac{k^3}{C_p^3Z^3}\right)\right]^{1/8}$$

in which

F = 0.27 for air at 68 F and atmospheric pressure
D = proportionality constant
k = thermal conductivity
C_p = specific heat
ρ = density
B = coefficient of thermal expansion
Z = viscosity
M = molecular weight
R = gas constant

Since B, M, and the ratio C_pZ/k (ref. 5, p. 417) are independent of P and T, we may write

$$\left(\frac{F}{C_pM^{1/2}}\right)^{1/2} = D'(\rho^2Z^2)^{1/8}$$

and ρ for a gas is MP/RT, then

$$\left(\frac{F}{C_pM^{1/2}}\right)^{1/2} = D'\left(\frac{M^2P^2}{R^2T^2}\cdot Z^2\right)^{1/8}$$

Again M and R are independent of P and T, so

$$\left(\frac{F}{C_pM^{1/2}}\right)^{1/2} = D''\left(\frac{P^2Z^2}{T^2}\right)^{1/8}$$

$$= D''\left(\frac{P^2T^{1.398}}{T^2}\right)^{1/8}$$

$$= D''\left(\frac{P^2}{T^{0.602}}\right)^{1/8}$$

$$\left(\frac{F}{C_pM^{1/2}}\right)^{1/2} = D''\left(\frac{P^{1/4}}{T^{0.0753}}\right)$$

Z^2 Varies with T for air as follows:

T	Z	Z^2
460 deg R	0.0162	0.0002625
528	0.018	0.000323
660	0.021	0.000440
860	0.025	0.000625
1060	0.029	0.000840
1260	0.032	0.001050
1460	0.036	0.001295
1560	0.038	0.001444

$$\left(\frac{1560}{460}\right)^n = (3.39)^n = \frac{1.444}{0.2625} = 5.50$$

$$n = 1.398$$

When P = 14.7 psia, and T = 528 deg R, C_p = 0.24 and F = 0.27 for air

$$\therefore D'' = \left(\frac{0.27}{0.24\times 29^{1/2}}\right)^{1/2}\times\frac{(528)^{0.0753}}{(14.7)^{1/4}}$$

$$D'' = 0.375$$

and

$$\left(\frac{F}{C_pM^{1/2}}\right)^{1/2} = 0.375\,\frac{P^{1/4}}{T^{0.0753}}$$

Now, it is also necessary to show that $\left(\frac{F}{C_pM^{1/2}}\right)^{1/2}$ gives higher values for air than for any other gases whose properties are available in the reference literature. To do this we use McAdams Equation [20],[25] again which is

$$F = D\left(\frac{\rho^2BC_pk^3}{Z}\right)^{1/4}$$

and in which the nomenclature is as before. Again, $\rho = MP/RT$ and $B = 1/492$. Then, at a constant pressure and temperature it would be expected that F would vary for different gases as follows:

$$F = D'\left(\frac{M^2C_pk^3}{Z}\right)^{1/4}$$

At 1 atm pressure and 20 C, F = 0.27 and $\left(\frac{M^2C_pk^3}{Z}\right)^{1/4}$ = 0.465 for air, so that the F value for any other gas, whose molecular weight, specific heat, thermal conductivity, and viscosity are known, may be determined from the relation

$$F_g = \frac{0.27}{0.465}\left(\frac{k^3C_pM^2}{Z}\right)_g^{1/4}$$

in which the subscript g applies to the gas other than air. On this basis, the F value for a number of representative gases has

[25] Ref. 5, p. 242.

TABLE 3 VALUE OF F FOR REPRESENTATIVE GASES

Gas	k^a	$C_p{}^a$	Z^a	M^a	$\frac{k^3C_pM^2}{Z}$	$\left(\frac{k^3C_pM^2}{Z}\right)^{1/4}$	F	$\frac{F}{C_pM^{1/2}}$
Air	0.0161	0.24	0.018	29	0.0468	0.465	0.270	0.209
Hydrogen	0.1075	3.50	0.009	2	1.9324	1.181	0.685	0.138
Acetone	0.0065	0.34	0.0077	58	0.0408	0.448	0.260	0.100
Acetylene	0.0124	0.43	0.010	26	0.0554	0.485	0.281	0.128
Benzene	0.0062	0.26	0.0075	78	0.0502	0.472	0.273	0.135
Ethyl alcohol	0.0089	0.40	0.0094	46	0.0634	0.501	0.290	0.107
Ethyl ether	0.0088	0.46	0.0077	74	0.222	0.687	0.398	0.100
Water	0.0110	0.48	0.0097	18	0.0213	0.382	0.221	0.108
Pentane	0.0083	0.34	0.0063	72	0.1599	0.632	0.366	0.127
Carbon dioxide	0.0109	0.21	0.0146	44	0.0360	0.436	0.252	0.181
Freon 113''	0.0054	0.16	0.0103	187	0.0855	0.542	0.313	0.143

[a] Ref. 5, various pages.

been calculated as shown in Table 3, and it will be noted that the value of $\left(\frac{F}{C_pM^{1/2}}\right)^{1/2}$ for air is greater than for any of those values so obtained; and for many others, such as ammonia, nitrogen, and oxygen, whose properties were considered but not listed in the table.

Use of General Equations [9] or [3]″, and [10] or [15]″

The following example is given of the use of Equation [3]″ for a 10 ft 6 in. diam (D) × 18 ft 10 in. high over-all vertical tank with dished heads, nominally full of ethyl ether, operating at 15 psig, and surrounded with intense fire

$$d = \sqrt{\frac{IS_w}{205.5(P + 14.7)}}\left(\frac{T}{Mr^2}\right)^{1/4}$$

$$d = \sqrt{\frac{20{,}000 \times 583}{205.5 \times 31.2}} \times 0.140$$

$$d = 5.96 \text{ in.}$$

in which

$S_w = \pi D(0.266\,D + L)$
L = height of straight shell to filling level, ft
$S_w = \pi\ 10.5(0.266 \times 10.5 + 14.9)$
$S_w = 583$ sq ft
$P = (16.5 + 14.7) = 31.2$
$\left(\frac{T}{Mr^2}\right)^{1/4} = 0.140$ (see Fig. 12)

An example of the use of Equation [15]″ for a 10 ft 6 in. diam (D) by 18 ft 10 in. high over-all air tank with dished heads, operating at 15 psig and surrounded by intense fire, is as follows

$$d = 0.02615\,\frac{S^{1/2}}{P^{1/4}} \cdot \frac{(T_w - T)^{5/8}}{T^{0.325}}$$

$$d = \frac{0.02615 \times 26.45 \times 10.85}{2.363}$$

$$d = 3.17 \text{ in.}$$

in which

$S = \pi D(0.532\,D + L)$
L = length of straight shell between heads, ft
$S = \pi\ 10.5(0.532 \times 10.5 + 15.67)$
$S = 700$ sq ft, $S^{1/2} = 26.45$
$T = (0 + 460) = 460$ F abs
$T_w = (460 + 1100) = 1560$ F abs
$(T_w - T)^{5/8} = (1100)^{5/8} = 79.5$

$$T^{0.325} = (460)^{0.325} = 7.33$$

$$(T_w - T)^{5/8}/T^{0.325} = 10.85$$

$$P = (16.5 + 14.7) = 31.2$$

$$P^{1/4} = 2.363$$

Simplification of General Expressions to Equations [7] or [16]″, and [8] or [17]″

The diameter of a connection necessary to relieve a pressure vessel in liquid service when involved with conflagration is as follows:

$$d = \sqrt{\frac{IS_w}{205.5\,P}}\left(\frac{T}{Mr^2}\right)^{1/4} \quad \text{[3]″}$$

$$d = \sqrt{\frac{20{,}000\,S_w}{205.5\,(P + 14.7)}} \times [0.130 + 0.0003\,(P + 14.7)]$$

$$d = \sqrt{\frac{97.3\,S_w}{(P + 14.7)}}\left(\frac{P + 448.0}{3333.3}\right) \quad \text{[16]″}$$

in which

I = 20,000 Btu per hr per sq ft
$\left(\frac{T}{Mr^2}\right)^{1/4}$ = contents factor, Fig. 12
S_w = wetted surface exposed, sq ft
P = 1.1 working pressure, psig

The diameter of a connection necessary to relieve a pressure vessel in gas or vapor (only) service when involved with conflagration is as follows

$$d = 0.02615\,\frac{S^{1/2}}{P^{1/4}} \cdot \frac{(T_w - T)^{5/8}}{T^{0.325}} \quad \text{[15]″}$$

$$d = 0.02615\left(\frac{S^{1/2}}{P^{1/4}}\right) \times 10.85$$

$$d = \sqrt{\frac{0.0803\,S}{\sqrt{(P + 14.7)}}} \quad \text{[17]″}$$

in which

S = surface exposed, sq ft
$\frac{(T_w - T)^{5/8}}{T^{0.325}}$ = *gas expansion variable,* as described under that heading, and as reduced to maximum constant (10.85) in preceding example
P = 1.1 working pressure, psig

Use of Equations [7] or [16]″, and [8] or [17]″

An example of the use of Equation [16]″ for a 10 ft 6 in. diam by 18 ft 10 in. high over-all vertical tank with dished heads, nominally full of volatile liquid, operating at 15 psig, and surrounded by intense fire is as follows

$$d = \sqrt{\frac{97.3\,S_w}{(P + 14.7)}}\left(\frac{P + 448.0}{3333.3}\right) = \sqrt{\frac{97.3 \times 583}{(16.5 + 14.7)}}\left(\frac{464.5}{3333.3}\right)$$

$d = 5.94$ in.

The following example gives the use of Equation [17]″ for a 10 ft 6 in. diameter by 18 ft 10 in. high over-all tank with dished heads in gas (only) service, operating at 15 psig, and surrounded by intense fire

$$d = \sqrt{\frac{0.0803\,S}{\sqrt{(P + 14.7)}}} = \sqrt{\frac{0.0803 \times 700}{\sqrt{(16.5 + 14.7)}}}$$

$d = 3.17$ in.

Simplification of Weight Rate to Vapor Flow Capacity in Terms of Area

Weight rate of flow

$$W = 262\,aP\sqrt{\frac{M}{T}} \text{ lb per hr} \quad \text{[1]″}$$

Flow capacity in terms of cubic feet of vapor per minute

$$Q \text{ cu ft} = \frac{W \text{ lb}}{\rho \text{ lb per cu ft}}$$

$$Q = \frac{262}{60}\,aP\left(\frac{M}{T}\right)^{1/2}\frac{10.72(528)}{M(14.7)}$$

$$Q = \frac{1682\,aP}{\sqrt{MT}} \text{ cfm of free contents vapor at 68 F} \quad \text{[18]″}$$

where

$$\rho = \frac{MP}{RT} = \frac{M(0 + 14.7)}{R(68 + 460)} = \frac{M(14.7)}{10.72(528)}$$

at 68 F and atmospheric pressure

Flow capacity in terms of cubic feet of air per minute:

$$W = \frac{0.533\,CAP}{\sqrt{T}} \text{ lb per sec of air}^{26}$$

$$Q \text{ cu ft} = \frac{W \text{ lb}}{\rho \text{ lb per cu ft}}$$

$$Q = \frac{0.533 \times 0.83 \times aP \times 60}{0.0752\,\sqrt{T}}$$

$$Q = \frac{353.0\,aP}{\sqrt{T}} \text{ cfm free air at 68 F} \qquad [19]''$$

where

C = orifice coefficient, 0.83

$$\rho = \frac{MP}{RT} = \frac{29(14.7)}{10.72(528)}$$

ρ = 0.0752 at 68 F and atmospheric pressure

Flow capacity in terms of pounds of saturated steam per hour: Napier's formula

$$W = \frac{aP}{70} \text{ lb per sec}$$

$$W = \frac{aP}{70} \times 3600 \times 0.83$$

(0.83 coefficient of flow for a short tube)

$$W = 42.7\,aP \text{ lb per hr of saturated steam} \qquad [20]''$$

Derivation and Use of Time Element Formulas

The estimated time required for the liquid contents of a vessel to reach a given temperature is derived as follows:

IS_w = total heat input, in Btu per hr

$\frac{V\rho C_p \Delta T}{\theta}$ = heat absorbed by liquid contents, Btu per unit time

$S \cdot \frac{t\rho C_p \Delta T}{12\,\theta}$ = heat absorbed by tank, Btu per unit time

$$IS_w = \frac{V\rho C_p(T - 70)^*}{\theta} + S \cdot \frac{t}{12} \cdot \frac{487 \times 0.12(T + 20)^*}{\theta}$$

$$(S_w \leq 0.85\,S)$$

$$\theta = \frac{V\rho C_p(T - 70)}{IS_w} + \frac{4.87\,t(T + 20)}{I} \qquad [21]''$$

In which

I = unit heat input rate, 20,000 Btu per hr per sq ft
S_w = wetted surface exposed to fire, sq ft
V = volume of liquid contents, cu ft
ρ = density, lb per cu ft (487 for steel)
C_p = specific heat (0.12 for steel)
ΔT = temperature difference, deg F
θ = time, hr
S = tank surface heated, sq ft
t = thickness of tank shell, in.
T = temperature of contents, deg F

The following example is given of the estimated time, Equation [21]", required for the contents to reach the boiling point in a 10 ft 6 in. diam by 18 ft 10 in. high over-all tank with dished heads, nominally full of ethyl ether, operating at 15 psig, and surrounded by intense fire (the same nomenclature is used as for the preceding group):

$$\theta = \frac{V\rho C_p(T - 70)}{IS_w} + \frac{4.87\,t(T + 20)}{I}$$

$$\theta = \frac{(10{,}200 \text{ gal}/7.48) \times 44.5 \times 0.552\,(134.6 - 70)}{20{,}000 \times 583} + \frac{4.87 \times 0.375\,(134.6 + 20)}{20{,}000}$$

$$\theta = 0.186 + 0.0141 = 0.2001 \text{ hr or } 12.0 \text{ min}$$

Estimated time required for an empty steel tank to reach the "critical" temperature of 1100 F (the same nomenclature is used as before)

$$IS = S \cdot \frac{t}{12} \cdot \frac{487 \times 0.12(T - 50)}{\theta}$$

$$I = \frac{4.87\,t(1100 - 50)}{\theta}$$

$$\theta = \frac{5114\,t}{0.8I} = 0.320\,t \qquad [22]''$$

During the period of heating to 1100 F, the input I will be found only about 80 per cent effective due largely to reradiation losses.

Origin of Formulas for Ranges of Pressure

From 15 to 60 psig working pressure

$$d = \sqrt{\frac{97.3\,S_w}{(P + 14.7)}\left(\frac{P + 448.0}{3333.3}\right)} \qquad [16]''$$

in which

$$d = \sqrt{\frac{a}{0.7854}}$$

$$S_w \leq 0.85\,S$$

$$d = \sqrt{\frac{97.3 \times 0.85\,S}{(16.5 + 14.7)}\left(\frac{16.5 + 448.0}{3333.3}\right)}$$

$$d = \sqrt{0.0515\,S}\,; \qquad \frac{a}{0.7854} = 0.0515\,S$$

$$a = 0.0405\,S \qquad [23]''$$

From 60 to 200 psig working pressure:

$$d = \sqrt{\frac{97.3 \times 0.85\,S}{(66 + 14.7)}\left(\frac{66 + 448.0}{3333.3}\right)}$$

[26] "Chemical Engineer's Handbook," by J. H. Perry, McGraw-Hill Book Company, Inc., New York, N. Y., 1934, Equation [21a], p. 710.

* Liquid contents are considered to have an average temperature of 70 F, and tank steel an average of 60 F, before exposure. The tank shell after heating is considered to be an average of 100 F hotter than the liquid contents, although this will vary somewhat with the compound contained.

$$d = \sqrt{0.0244S}; \qquad \frac{a}{0.7854} = 0.0244S$$

$$a = 0.0192\ S \ldots\ldots [24]''$$

From 200 to 700 psig working pressure

$$d = \sqrt{\frac{97.3 \times 0.85S}{220 + 14.7}\left(\frac{220 + 448.0}{3333.3}\right)}$$

$$d = \sqrt{0.01415\ S}; \qquad \frac{a}{0.7854} = 0.01415\ S$$

$$a = 0.0112\ S \ldots\ldots [25]''$$

Example of Use of Final Formulas for Relief Requirements

Relief requirements for a 10 ft 6 in. diam (D) by 18 ft 10 in. high over-all tank with dished heads, in gas service, operating at 150 psig are determined as follows (equation numbers refer to Summary for Use):

Relief-connection size

$$a = 0.0192\ S \text{ or } d = \sqrt{0.0244\ S} \ldots\ldots [2]'$$

$$a = 0.0192 \times 700 \text{ or } d = \sqrt{0.0244 \times 700}$$

$$a = 13.44 \text{ sq in. or } d = 4.13 \text{ in. (use 4 in. ASA150)}$$

where

S = tank surface, sq ft
$S = \pi D(0.532\ D + L)$
L = length straight shell, ft
$S = \pi\ 10.5(0.532 \times 10.5 + 15.67)$
S = 700 sq ft

Relief capacity in terms of area:

$$a = \frac{0.0631\ S}{\sqrt{(P + 14.7)}} \text{ or } d = \sqrt{\frac{0.0803\ S}{\sqrt{(P + 14.7)}}} \ldots\ldots [5]'$$

$$a = \frac{0.0631 \times 700}{\sqrt{(165 + 14.7)}} \text{ or } d = \sqrt{\frac{0.0803 \times 700}{\sqrt{(165 + 14.7)}}}$$

$$a = 3.30 \text{ sq in. or } d = 2.047 \text{ in.}$$

Relief capacity of the device in terms of fluid flow

$$Q = \frac{353.0a\,P}{\sqrt{T}} \ldots\ldots [10]'$$

$$Q = \frac{353 \times 3.30\ (165 + 14.7)}{\sqrt{666}}$$

Q = 8100 cfm of air at 68 F, and atmospheric pressure

in which the temperature required to open valve, air having initial temperature of 150 F, is

$$T/P = T'/P'$$

$$T = \frac{179.7 \times (150 + 460)}{164.7}$$

$$T = 666 \text{ F abs}$$

[Provided the relief capacity required for operation fluctuations and disturbances (capacity of the compressor at 165 lb discharge) is not greater. Three and four-inch valves of this capacity are commercially available.]

PART 3 REQUIREMENTS FOR RELIEF OF ATMOSPHERIC VESSELS[27]

Introduction

A considerable number of references in the literature specifying the relief requirements of atmospheric storage tanks have been available for many years. Various methods have been developed to establish the normal venting requirements, and at least three nationally known, reliable organizations have agreed upon and proposed venting capacities necessary to limit internal pressure safely in event of exposure to fire. The fire records of the National Fire Protection Association and similar bodies clearly indicate the need and value of emergency vent areas for this contingency and emphasize the inadequacy of normal relief apparatus when a vessel is involved with intense fire. The inadequacy of the latter will be very apparent in this presentation.

In this section a recent investigation of the normal requirement made by the authors will be set forth and compared with previous methods. By the procedure used in Part 2 for pressure vessels, the emergency requirement will be developed, compared, and discussed. Thereafter, a summary for the suggested employment of this information will be given.

Summary

To summarize briefly the principal results of Part 3, Fig. 14 has been included, which shows the relation and suggested combination of normal and emergency vent sizes as determined for vertical cylindrical atmospheric storage tanks.

Study of Fig. 14, practical experience with present-day venting apparatus, and consideration for economy led to the venting requirements given in Table 7, which are considered adequate for both normal (1 in. water external, 3 in. water internal, pressure) venting and vaporization (at 0.5 psi at 50 gal, proportionally reduced to 3 in. water pressure at 70,000 gal) of contents due to fire exposure. The formulas from which the foregoing were calculated appear under the descriptive headings for which they apply. Other formulas which provide relief specifications for all vessels in liquid or gas service at working pressures not to exceed 0.50 psig are also given.

The suggested application of the information herein will be found under the heading, Proposed Summary for Use.

Thermal Breathing or Normal Venting of Uninsulated, Above Ground, Atmospheric Storage Tanks When Exposed to Extreme Atmospheric Changes

Introduction. During the years 1939 and 1940, for a period of 14 months, a continuous record of breathing characteristics was kept on a 440,000-gal (nominal capacity) gasoline storage tank at South Charleston, West Va., (38 deg north latitude, 600 ft elevation). Rates of venting, specific gravities of the vapors, vapor pressures, liquid levels, liquid temperatures, vapor temperatures, and atmospheric temperatures were measured and recorded. The data obtained furnished information for vapor-conservation analysis (the subject of a separate discussion) and the determination of normal breathing requirements.

Summary. Maximum thermal breathing was found to be that due to contraction during a hailstorm. A temperature drop of 41 deg F (average) in 20 min was observed within the tank vapor. The inhaling rate resulted in vent-size requirements larger than those determined by a number of well-known methods, except those sizes proposed by the National Board of Fire Underwriters[28] (up to 800,000 gal capacity).

[27] Conditions: 1 in. water external, 0.50 psi internal, pressure.
[28] "Containers for Storing and Handling Flammable Liquids," National Board of Fire Underwriters, Pamphlet No. 30, New York, N. Y., 1941, p. 37.

It was concluded that the maximum rate of thermal breathing of an atmospheric (1 in. water external, 3 in. water internal, pressure) storage tank is

$$Q = (S/0.290) + \text{emptying rate} \qquad [16]$$

where Q and emptying rate are in cubic feet of air per hour at 68 F and atmospheric pressure, and S is the area of the tank roof and shell in square feet.

The diameter of the relief connection required for this venting rate is

$$d = \sqrt{Q/921} \qquad [17]$$

where d is in inches and Q is in cubic feet of air per hour at the conditions given. Derivation of these formulas will be found in the Appendix of this section.

In order that the vent sizes determined by Equations [16]

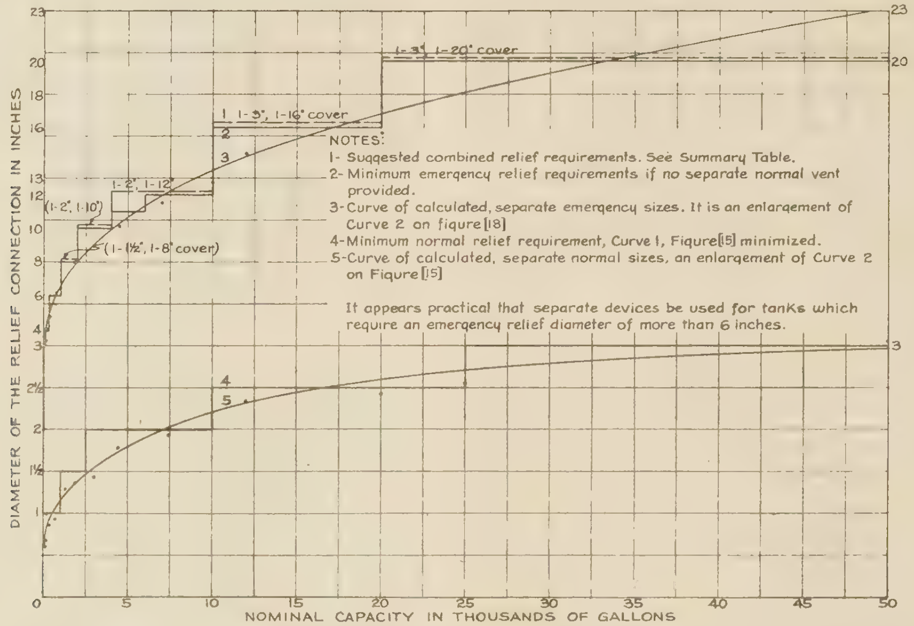


Fig. 14 Comparison of Normal and Emergency Relief Areas for Atmospheric Tanks to Facilitate Selection of Practical and Economical Combinations of the Two

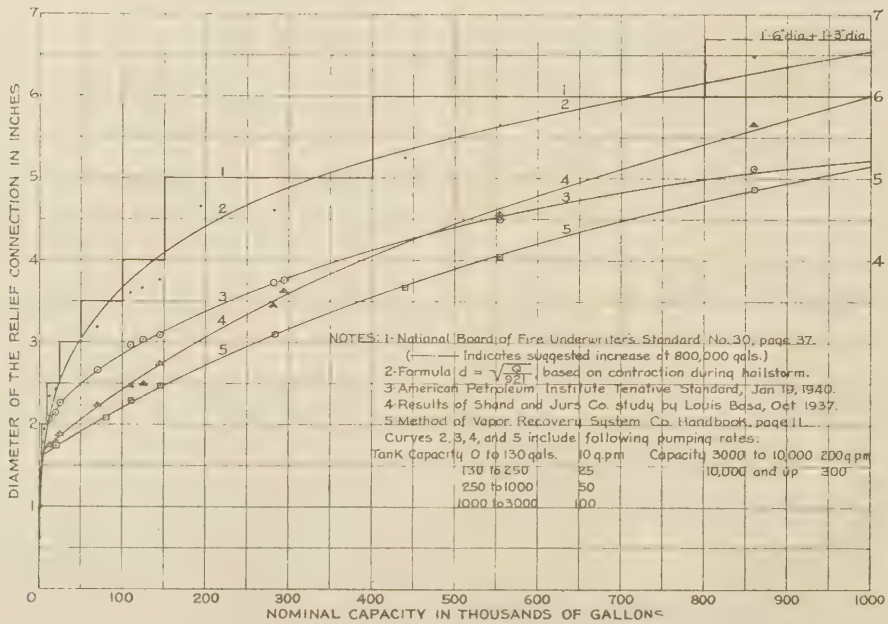


Fig. 15 Comparison of Required Diameter of Relief Connection for Atmospheric Pressure Storage Tanks Exposed to Extreme Atmospheric Changes

(1 in. water external, 3 in. water internal, pressure.)

and [17] may be compared with those specified by other methods, Fig. 15 is given.

Equipment Arrangement. The tank on which a continuous record was kept was 50 ft diam × 30 ft 3 in. high (cylindrical shell), with an umbrella-type roof, and was painted with aluminum paint. The vessel was in active storage of finished gasoline.

An existing 8-in. standard pipe header about 600 ft long was used as a vent line. Two recording orifice meters were installed near the end of the line to measure both high and low flows in either direction. Also in connection therewith, a continuous-recording gravitometer was used to provide data for the vapor-density correction. A three-pen recording thermometer furnished atmospheric, tank vapor, and liquid-contents temperatures.

At the same time, other data were collected pertinent to related studies. Periods when the contained vapors were flammable were recorded by a combustible-gas instrument and recording potentiometer. While the effects of external water spray on vapor losses were being observed, a third orifice meter was used to measure the cooling-water rate. Frequent periodic analyses of the discharge vapors were made to determine the percentage of product contained at the indicated specific gravities.

Procedure. The experiment was given continuous attention and maintenance by operation and investigation personnel while the tank was being operated normally. A daily log was kept on all observations including weather, liquid level, vapor pressure, and pertinent comment. The data were transposed to a graph, a sample of which is included as Fig. 16. On this will be noted both ordinary and extreme breathing action during three representative periods.

Vapor and liquid temperatures and flammability of vapors were observed at various levels to obtain mean temperatures and to get indication of the location, frequency, and extent of combustible zones. Flash-arrester and emergency vacuum and pressure-relief protection, were maintained on the tank throughout the observations.

Data Obtained. The maximum breathing rates and corresponding information observed during the 14 months under various conditions are shown in Table 4. The contraction rate due to sudden cooling will be seen to be about 4 times as great as that for emptying, filling, solar heating, or combinations of conditions causing venting. No worth-while data on the efflux produced by mixing (by cycling contents) could be obtained other than that breathing rates so caused are not comparable to the maximum shown in Table 4.

Although not directly related to the subject, the data collected on the flammability of the vapor-air mixture were of interest. To show the relation between combustible contents and breathing action, Fig. 17 has been included.

For the most part (except for 3 months) the combustible-gas recorder operated in connection with the vent header at a point near its outlet. Flammable mixtures were present there at frequent intervals in an irregular pattern which was no doubt due to the lag in and large volume of the 600-ft long header. The interval from February 15, through February 26, 1940, Fig. 17, illustrates flammability within the header.

During June, July, and August, 1940, the combustible-vapor recorder operated directly in connection with the tank. It was found that explosive mixtures were present less frequently but for longer periods ranging up to 30 hr. In all, the tank vapors were within the flammable range 5 per cent of the time and "too rich to burn" 95 per cent of the time. The vapor-air mixtures when combustible were never in the lower limits, the gasoline content ranging from 2 to 53 per cent by volume, as based on laboratory analyses. Flammable limits of gasoline vapor-air mixtures are 1 to $6^1/_2$ per cent by volume.[29]

All the recorded periods (seven in number) when the vapors within the tank were flammable are shown on the remainder of Fig. 17. It will be observed that in every case (except one of 45 min on June 17) the combustible mixture was produced by a combination of emptying and falling temperature. The exception was caused by a momentary in-breathing when a shower occurred while the vapors were warm (between 9 and 10 a.m.).

The use of flash arresters or inert-gas systems with atmospheric storage tanks containing volatile flammable liquids appears warranted by these observations.

Analysis. The data indicated that the maximum normal breathing requirement is that due to rapid contraction of the contained vapors. The quantity rate of contraction of a gas due to heat transferred to a colder tank wall is

$$Q = \frac{10.72FS(T - T_w)^{5/4}}{C_pMP} \quad \ldots\ldots\ldots\ldots [18]$$

in which Q is in cubic feet per hour, F is a function of the physical properties of the vapor (0.27 for air at atmospheric conditions),[30] S is the surface of the tank roof and shell enclosing the tank vapor in square feet, T is the initial vapor temperature in deg F, T_w is the final tank wall temperature (average) in deg F, C_p is the specific heat of the vapor, M is the molecular weight, and P is the average absolute pressure during contraction in pounds per square inch (14.736 internal for atmospheric tanks).

If the properties of air which constituted the greater percentage of the mixture ordinarily, and nearly the whole occasionally (air ranged 47 to 98 per cent by volume), and the observed temperatures from Table 4, Case 1, are used in Equation [18], maximum values of Q will be obtained. (For comparison with values for other vapors, see the evaluation of $\left(\frac{F}{C_pM^{1/2}}\right)^{1/2}$ in the Appendix of this section.) This value (cubic feet per hour) is then very nearly equal to $S/0.290$ as in Equation [16].

The capacity of the relief connection (a short tube) at pressures in this range is

$$Q = 4340\, d^2 \sqrt{\frac{\Delta PT}{MP}} \quad \ldots\ldots\ldots\ldots [19]$$

in which Q and M are the same as in Equation [18], d is in inches, ΔP is pounds per square inch gage, P is pounds per square inch absolute, and T is the temperature of the vapor flowing in deg F absolute.

During the cooling process, the capacity of the relief connection must be equivalent to the rate of contraction at the permissible working pressure (1 in. water external). So, if Equation [18] is equated to Equation [19] at the observed conditions (Case 1), the required diameter will result as in Equation [17] $d = \sqrt{Q/921}$.

Derivation of Equations [18] and [19] will be found in the Appendix of this section.

Discussion. Although the contained vapors are not perfect gases and some condensation does take place during cooling, the comparison between observed breathing rates and those computed as for a gas on the basis of the rate of change of absolute temperatures is favorable and indicates results in the direction of safety.

The venting rate, $Q = S/0.290$, is based on a difference between initial vapor and final average tank-wall temperatures of 47 deg F and is equivalent to a fall in the vapor temperature of

[29] Ref. 9, p. 152.
[30] "Heat Transmission,"[5] p. 242.

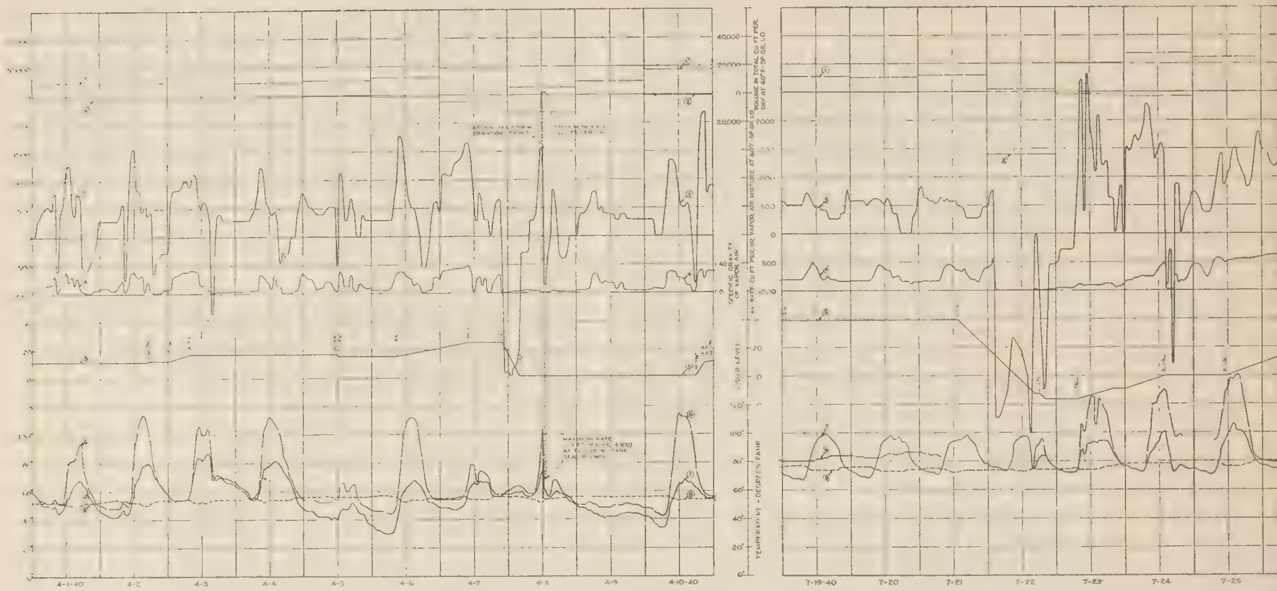
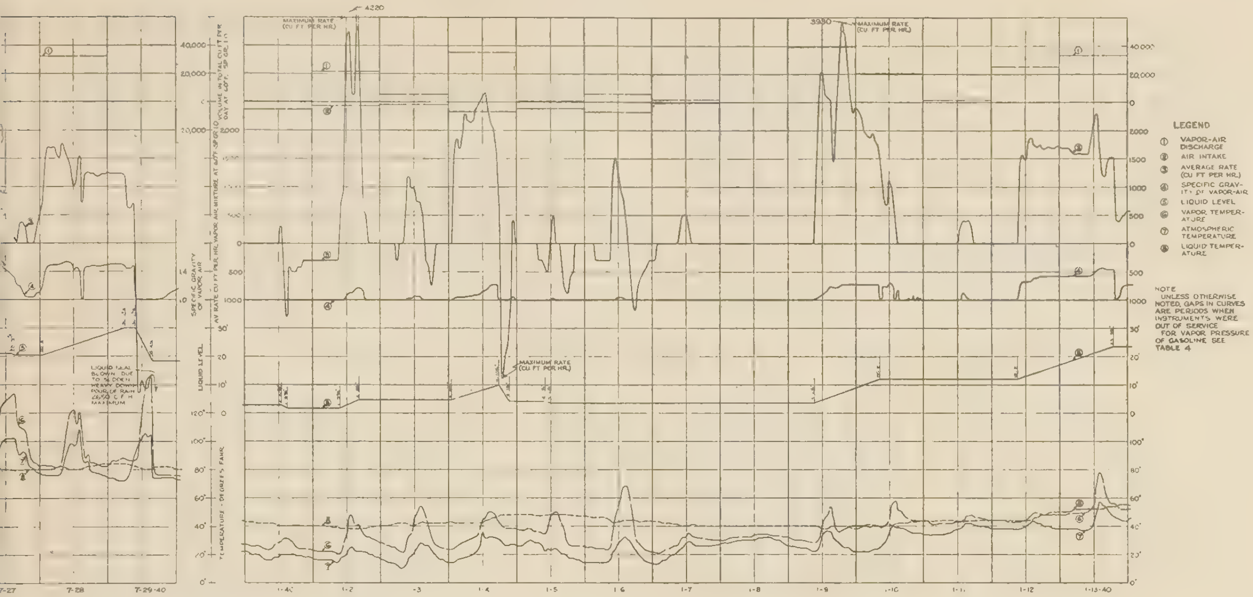
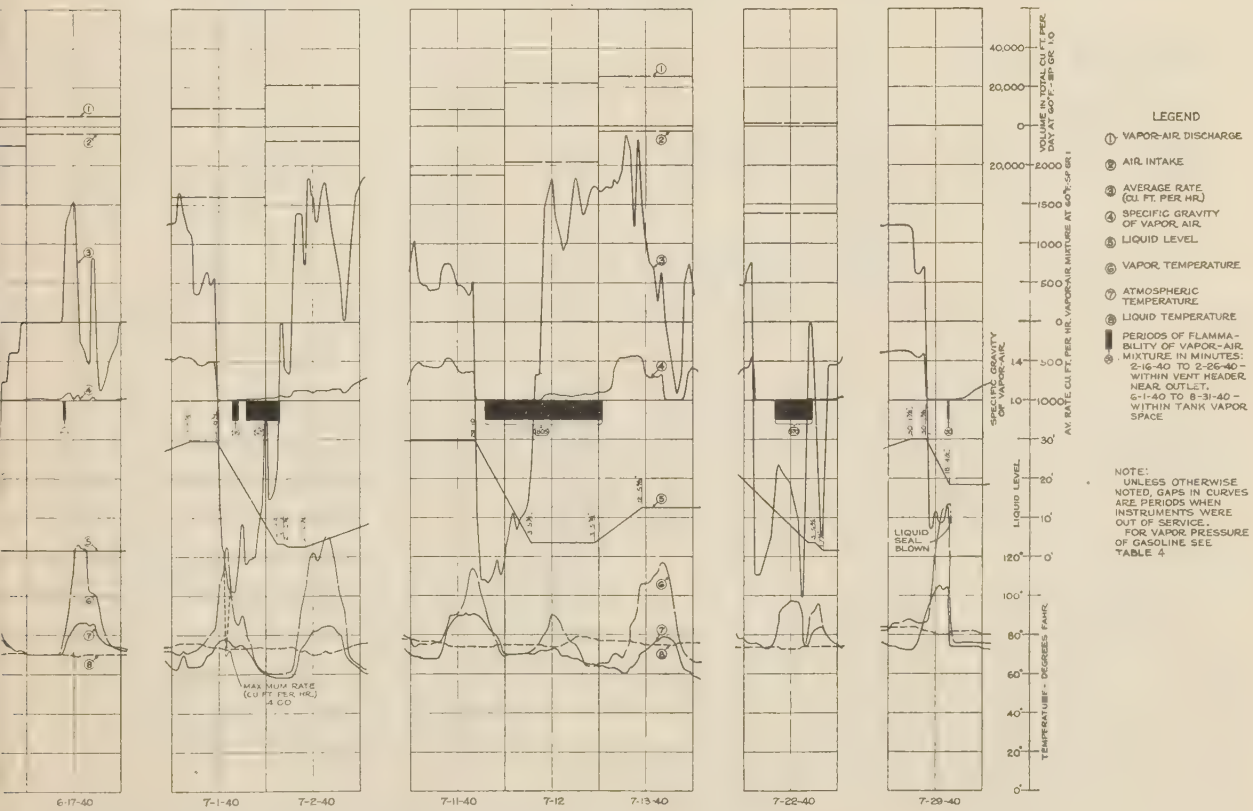


FIG. 16 BREATHING CHARACTERISTICS OF 10,000-
(Significant periods of 14 months' continuous re

FIG. 17 RELATION OF FLAMMABILITY OF TANK VAPORS TO BREATHING CHA
(Representative peric



ge Tank Containing Finished Gasoline
wn in Table 4, Part 3 of this paper.)



tics of 10,000-Bbl Storage Tank Containing Finished Gasoline
nths' observation.)

TABLE 4 OBSERVED AND CALCULATED DATA ON MAXIMUM ... AGE TANK WHEN EXPOSED TO

Conditions — Temperatures:	Hail Storm April 8, 1940 11:50 a.m. to 12:10 p.m. Case 1	Sudden Showers July 29, 1940 3:40 p.m. Case 2	May 31, 1940 2:05 p.m. Case 3
Atmospheric	73° to 54°F. in 20 min.	104° to 74°F. in 45 min.	78° to 65°F. in 30 min.
Vapor Contents	103° to 62°F. in 20 min.	146° to 84°F. in 1 hr.	95° to 67°F. in 30 min.
Maximum rate of change (vapor)	98° to 70°F. in 7 min.	119° to 111°F. in 2 min.	92° to 74°F. in 15 min.
Liquid Contents	52° to 56°F. in 20 min.	80° to 82°F. in 1 hr.	58°F. constant
Final Tank Wall about Vapor Space	56.0°F. (estimated average)		
Recorded Vapor Flow Rate (Max.)	4300 cu.ft. per hr. when 1 1/2 oz. per sq. in. vacuum seal blown	2650 cu.ft. per hr., when seal blown	*4450 cu.ft. per hr.
Liquid Level	10'-7 5/8" (constant)	19'-0" (constant)	0'-9" (constant)
Vapor Space Quantities	Volume 45,200 cu. ft. Surface (S) 5170 sq. ft.	Volume 28,800 cu.ft. Surface(S) 3860 sq.ft.	Volume 64,600 cu.ft. Surface(S) 6720 sq.ft.
Rate of Volume Change (Proportional to maximum abs. temp. rate as a gas).	18,850 cu. ft. per hr.	12,000 cu.ft. per hr.	*8400 cu.ft. per hr.
Proposed Breathing Rate (Based on S/0.290 as derived. Pump rate to be added).	17,830 cu. ft. per hr.	13,300 cu.ft. per hr.	23,200 cu.ft. per hr.
Reid Vapor Pressure (lb. per sq. in. at 100°F.)	8.3 lb. per sq. in.	8.3 lb. per sq. in.	5.6 lb. per sq. in.
Weather	Bright sun preceding storm, cloudy thereafter.	Hot sun shining before & after heavy rain storm.	Cloudy with intermittent sun before shower; bright sun thereafter.
Remarks:	Maximum condition (largest temperature drop in appreciable period found) used as basis of S/0.290.		

* Discrepancy between observed breathing rates and proportional rates of volume change is considered largely due to the pressure drop and lag in the ... recorded vapor flows were corrected to 60 F, the maxima at flowing

98 F to 70 F, in 7.86 min, when the vapor is assumed to be a gas. No additional temperature differential is considered necessary for a margin of safety. The vent sizes resulting are shown in Fig. 15, curve 2, which have a satisfactory service record of a number of years over a broad section of the country. As a matter of fact, a large number of comparable atmospheric tanks so vented were exposed to the same hailstorm cited, without giving any evidence of having been subjected to pressure above 1 in. of water external.

The capacity of a short tube, Equation [19], was derived from the adiabatic flow of an ideal gas through an orifice. It incorporates an orifice coefficient of 0.70 for flows where the ratio of upstream to downstream absolute pressure is less than 1.05.[31] Also, this is the coefficient used by the National Fire Protection Association under these conditions.[32] Tests were run on a typical tank and connection as a check on the formula derived. Actual flows were found to be 99 per cent of the theoretical within the atmospheric tank pressure range (0 to 3 in. water) and to be 100 per cent at a pressure of 5 in. of water (see Fig. 20 in the Appendix of this section).

[31] "Kent's Mechanical Engineers' Handbook," ref. 14, pp. 1–11.
[32] "Handbook of Fire Protection,"[9] p. 108.

Conclusion. The conclusions as outlined under *Summary* are adequate for the conditions specified and conform to the proposals of the National Board of Fire Underwriters up to 800,000 gal capacity, above which increased vent sizes as shown are suggested. Apparently, the average practice is to use vent sizes considerably smaller than this service-proved proposal (refer to Fig. 15).

Proposed sizes of connections with minimum capacities (based on 1 in. water external pressure) for the normal relief requirement for atmospheric tanks are listed in Table 5.

Relief capacities, concluded in Table 5, are not adequate for the condition of fire exposure now to be discussed and under which the relation of normal and emergency venting will be clarified.

To avoid excessive internal pressure due to accidental overfilling, the normal relief area should not be less than that of the fill pipe.

Requirements for Emergency Relief Due to Fire Exposure

Introduction. From the standpoint of emergency venting, atmospheric tanks (usually large) present a separate problem because of the very low safe working pressure (about 3 in. water

CHING RATES OF 10,000-BBL FINISHED GASOLINE STOR-
EME ATMOSPHERIC CHANGES

ar Heating , 1940 - 7 hrs. Case 4	Emptying May 1, 1940 - 6-1/3 hrs. Case 5	Emptying April 19, 1940 - 5 hrs. Case 6	Filling Jan. 2, 1940 - 7 hrs. Case 7	Filling Jan. 9, 1940 - 23 hrs. Case 8
74°F. in 7 hrs.	73° to 61°F. in 30 min.	58° to 57°F. in 1 hr.	28°F. constant	31° to 26°F. in 5 hrs.
112°F. in 6 hrs.	78° to 62°F. in 45 min.	60° to 58°F. in 1 hr.	56°F. constant	33° to 32°F. in 4 hrs.
82°F. in 1 hr. 10 a.m.)	78° to 64°F. in 30 min.	As above	None	As above
constant	55° to 57°F. in 10 min.	55°F. constant	42°F. constant	40°F. constant
cu.ft. per hr.	*4275 cu.ft. per hr.	*4050 cu. ft. per hr.	4220 cu. ft. per hr.	3930 cu. ft. per hr.
3/8" (constant)	11'-7 3/4" to 3'-10 3/4" in 6 1/3 hrs. Avg. 1.225 ft./hr.	29'-5" to 17'-4" in 5 hrs. Avg. 2.42 ft. per hr.	1'-9 3/4" to 4'-8 1/2" in 7 hrs. Avg. 0.414 ft. per hr.	3'-6 1/2" to 12'-2 1/2" in 23 hrs. Avg. 0.380 ft. per hr.
63,100 cu.ft. ce(S) 6600 sq.ft.	Volume 44,400 cu.ft. Surface(S) 5110 sq.ft.			
cu.ft. per hr.	2200 contraction 2400 liq. discharge *4600 cu.ft. per hr.	Liquid Discharge Rate *4750 cu. ft. per hr.	Liquid Charging Rate 813 cu. ft. per hr.	Liquid Charging Rate 740 cu. ft. per hr.
0 cu.ft. per hr.	17,600 cu.ft. per hr.			
b. per sq. in.	9.5 lb. per sq. in.	8.7 lb. per sq. in.	11.3 lb. per sq. in.	12.6 lb. per sq. in.
& sunny all day.	Cloudy with intermittent rain all day.	Light rain all day.	Weak sun all day.	Blanket of snow on tank partly melted by a weak sun.
	Contraction was due to a sudden shower during first 30 min.	Contraction due to temperature drop is small enough to be neglected.	Vaporization promoted by turbulence and contained air caused larger discharge.	Out breathing is again about five times greater than displacement.

ne, 600 ft in length, 8 in. diam. (It is also possible that the liquid seal may have been giving partial relief.) Furthermore, although the maximum
ons were not materially larger than the corrected flows.

TABLE 5 PROPOSED SIZES OF VENT CONNECTIONS FOR ATMOSPHERIC TANK NORMAL RELIEF REQUIREMENTS

Tank capacity, gal	Relief size, in. I.P. size	Minimum capacity, cu ft of air, per hr at 60 F, and atmospheric pressure
0 to 500	1[a]	708
500 to 2000	1 1/2[a]	1920
2000 to 10,000	2	3660
10000 to 50000	3	8230
50000 to 150000	4	14620
150000 to 400000	1–4, 1–3[b]	22820
400000 to 800000	6	32900
800000 to 1100000	1–6,1–3	41100
1100000 to 1300000	1–6, 1–4[c]	47500
1300000 to 1800000	8	58450
1800000 to 2000000	2–6	65800

[a] Units of 2-in. vent size for tanks are usually the smallest standard manufacture.
[b] One 6-in. vent unit may be cheaper than the two small units, but they have the mechanical advantage of multiple devices.
[c] Similarly, one 8-in. vent unit may be cheaper than one 6-in. and one 4-in. size.

internal), the relatively large vapor space, the lessened possibility of complete envelopment in flame, and the lengthened (time) exposure required to produce maximum vaporization. The effect of the first two of the factors mentioned is such as to result in calculated relief areas which are impractical. The latter two are the basis of reasonable assumptions to be made which result in more economical designs.

Summary. The required diameter of a relief connection to limit internal pressure (not to exceed 0.50 psig) within vessels when the volatile contents are absorbing heat at 20,000 Btu per sq ft per hr of wetted surface exposed is

$$d = \sqrt{\frac{0.913\, S_w}{\sqrt{P(P + 14.7)}}} \qquad [20]$$

If the pressure limit be 3 in. of water, as for atmospheric storage tanks, Equation [20] becomes

$$d = \sqrt{0.721 S_w} \qquad [21]$$

and if the atmospheric tank be vertical cylindrical with bottom resting on pad or foundation, Equation [21] becomes

$$d = \sqrt{2.265\, DH} \qquad [22]$$

in which (Equations [20], [21], and [22]) d is the internal diameter of the connection in inches, S_w is wetted surface exposed to fire in square feet when nominally full, P is the pressure limit

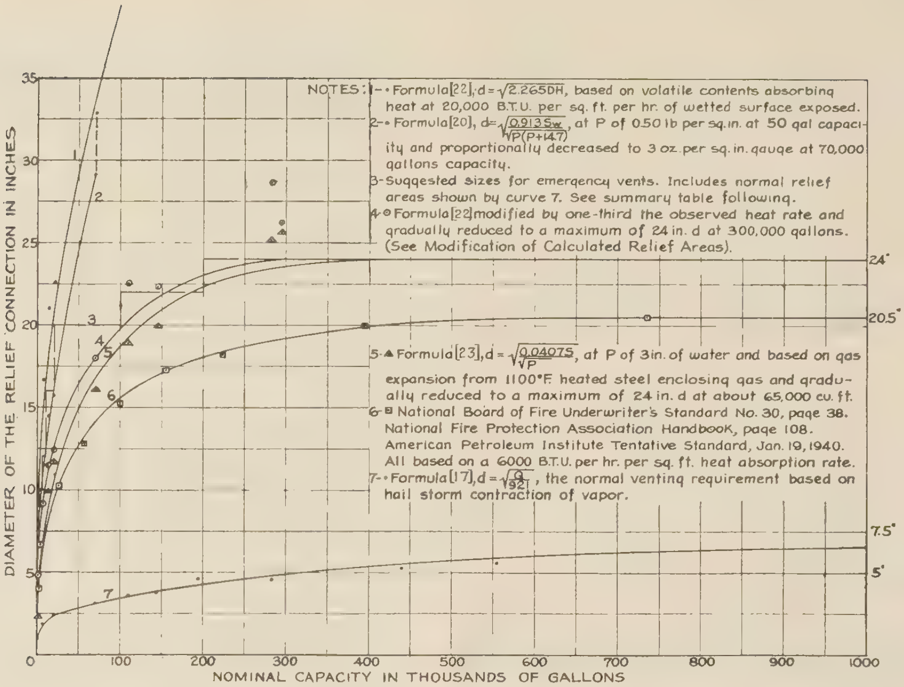


Fig. 18 Comparison of Required Diameter of Relief Connection for Atmospheric Storage Tanks When Exposed to Fire
(1 in. water external, 3 in. water internal, pressure.)

in pounds per square inch gage (0.1083 for 3 in. water), D is the diameter of the tank in feet, and H is the height in feet of the contents on the exposed tank wall when it is nominally full.

For vessels in gas service only at pressures not to exceed 0.50 psig, the required diameter of the relief connection to limit internal pressure to a specified maximum is

$$d = \sqrt{\frac{0.0407\,S}{\sqrt{P}}} \quad \text{[23]}$$

in which d is the internal diameter of the connection in inches, S is the square feet of surface which may be exposed including unprotected roofs, and P is the maximum pressure permitted in pounds per square inch gage.

The origin of these formulas will be found in the Appendix of this section.

The formulas for liquid service, Equations [20], [21], and [22], should be applied to all vessels in the pressure range below 0.50 psig up to and including 15 ft diam; it then appears expedient for practical reasons to reduce the calculated relief sizes and to limit them to a maximum of 24 in. diam. These modifications are clarified as the discussion proceeds and are summarized under the heading, *Modification of the Calculated Relief Areas.*

Relief diameters for atmospheric tanks have been graphed in Fig. 18. The curve of Equation [22] is shown in relation to the same equation when modified by a heat rate one third of that concluded in Part 1. Between the two will be noted the suggested choice of vent size. For comparison, the proposal of the National Board of Fire Underwriters, the National Fire Protection Association, and the American Petroleum Institute is shown. Also appearing is the required size for normal venting which, though provided for in the emergency area, is not large enough to reduce it one pipe size over the entire range of tank capacity. Also observe that a contained gas or vapor will expand when in relatively large proportions, as in empty atmospheric tanks and when exposed to tank walls heated to 1100 F by external fire, at a rate nearly equivalent to the vaporization rate of volatiles.

Analysis and Method. The requirements for emergency relief of atmospheric tanks and low-pressure gasholders are based on the same reasoning as that for unfired pressure vessels as detailed in Part 2 of this paper. The physical conditions (pressure, temperature, and relative size) are merely different, necessitating the use of a different fluid-flow formula, a different value for the contents factor, and modification of calculated relief sizes as graphically suggested in Fig. 18.

As previously described, a vessel surrounded by fire absorbs heat and the temperature of its contents rises. If the content is liquid, its vapor pressure steadily increases and eventually boiling will occur; if a vapor or gas, progressive expansion takes place. (So-called empty tanks, equipped with adequate normal venting apparatus, have been observed to fail when involved with sudden, intense fire, although ignition of contents did not occur until after rupture.) Now, in either case, the relief apparatus must be of capacity equivalent to the rate of vaporization or the rate of volume increase. When the weight rate of the relief connection (at these low pressures) is equated to the weight rate of vaporization, the following Equation [24] results, which is the origin of Equations [20], [21], and [22]:

$$d = \sqrt{\frac{IS_w}{405\sqrt{P(P+14.7)}}\left(\frac{T}{Mr^2}\right)^{1/4}} \quad \text{[24]}$$

When the weight rate of expansion of a gas or vapor is equated to the weight rate of the relief connection, a second expression, Equation [25], results, which is the origin of Equation [23], namely

$$d = 0.0186 \cdot \frac{S^{1/2}}{\Delta P^{1/4}} \cdot \frac{(T_w - T)^{5/8}}{T^{0.325}} \quad \text{[25]}$$

In these equations, d is the internal diameter of the connection in inches, I is the heat-absorption rate in Btu per square foot per hour, S_w is wetted surface exposed in square feet, ΔP is the maximum pressure in pounds per square inch gage (0.1083 for 3 in. water), and S is total surface exposed in square feet. The terms $\left(\frac{T}{Mr^2}\right)^{1/4}$ and $\frac{(T_w - T)^{5/8}}{T^{0.325}}$ are the contents factor and gas expansion variable, which are reduced to constants, as described under subsequent headings. Equations [24] and [25], which are similar to Equations [8] and [9] in Part 2, are derived and simplified in the Appendix of this section to the useful Equations [20], [21], [22], and [23].

As in Part 2, the relief capacity is proposed in terms of area of the relief connection, because the designer is primarily interested in selecting an adequate fitting of standard size for the vessel to be constructed, and because of the variation in capacities of commercial relief apparatus. The vent apparatus may then be selected of a capacity, based on approved flow test, to pass the flow capacity of the calculated area, as will be shown.

Weight Rate of a Relief Connection. The vapor capacity of a short tube in pounds per hour at pressures not exceeding 0.5 psig is

$$W = 405\, d^2 \sqrt{\frac{MP(P + 14.7)}{T}} \quad \ldots\ldots\ldots\ldots [26]$$

in which W is the weight rate in pounds per hour, d is the internal diameter of the connection in inches, M is the molecular weight of the fluid, P is the inlet pressure in pounds per square inch gage, and T is the temperature of the fluid in deg F absolute.

This is the same formula used previously in this section for the determination of the normal breathing requirement, except that it was there given in cubic feet per hour. As stated before, it was shown to be accurate, and its derivation found in the Appendix of this section will be seen to conform to the procedure outlined by the American Gas Association.[33]

Contents Factor F, $\left(\frac{T}{Mr^2}\right)^{1/4}$. This term is identical with that derived and fully explained in Part 2 of this paper under the same heading. It is, as before, that which relates the properties of liquid contents. Since the developments in this section are limited to the pressure range of atmospheric to 0.50 psig, a single numerical value of the factor may be selected which would be adequate for any compound to be contained at these pressures.

For the present development, the value $F = 0.136$ has been used, for which reference is made to Fig. 12, Part 2. This value is considered adequate for any liquid which may be contained at these pressures, other than carbon tetrachloride. For this compound, use 0.155 for the contents factor in Equation [24].

Gas Expansion Variable, $\frac{(T_w - T)^{5/8}}{T^{0.325}}$. The rate of volume increase is dependent upon the rate of temperature rise within the vapor contents. Again, this is proportional to the value of the gas-expansion variable. If the formerly assumed maximum temperature difference of 1100 deg F, between initial vapor and final tank-wall temperatures is considered reasonable and adequate, the term reduces to the constant 10.85 (refer to Part 2, under the same heading as in this section).

As will be seen in the Appendix of this section, the variable was based upon the properties of air which apparently gives maximum values as therein demonstrated.

Flow Capacity in Terms of Area. If the relief diameters (not the next larger pipe size) as determined above, the molecular weight of contents, the relieving pressure, and the vapor temperature, are inserted in the weight rate of flow, Equation [26], of a short tube, the necessary capacity of the relief device may be computed. The apparatus may then be selected of a capacity (based on approved flow test) to pass the calculated rate which is as follows:

$$W = 405\, d^2 \sqrt{\frac{MP(P + 14.7)}{T}} \text{ lb per hr of vapor} \ldots [26]$$

$$Q = 156{,}000\, d^2 \sqrt{\frac{P(P + 14.7)}{MT}} \text{ cfh of free vapor at 68 F} \ldots [27]$$

$$Q = 29{,}000\, d^2 \sqrt{\frac{P(P + 14.7)}{T}} \text{ cfh of free air at 68 F} \ldots [28]$$

In Equations [26], [27], and [28], d is in inches, P is in pounds per square inch gage, and T is in deg F absolute.

For conditions of fire exposure, T is approximately the normal boiling point of the liquid contents. For vessels in gas service, T is about the temperature necessary to raise the pressure in the vessel from the absolute working pressure and temperature to the absolute set pressure of the relief device, as computed by the gas law, $T/P = T'/P'$.

A mathematical clarification of these equations is given in the Appendix of this section.

Time Element. The time required when exposed to intense fire for the liquid contents of a steel vessel to reach a specified average temperature may be estimated by applying the formula

$$\theta = \frac{V\rho C_p(T - 70)}{IS_w} + \frac{4.87t(T + 20)}{I} \quad \ldots\ldots\ldots [29]$$

in which θ is time in hours, V is the volume of contents in cubic feet, ρ is the density in pounds per cubic foot, C_p is the specific heat, T is the specified temperature in deg F, I is the heat-absorption rate in Btu per square foot per hour, S_w is the surface wetted by the product and exposed to fire in square feet, and t is the thickness in inches of the (steel) tank wall. This was introduced and derived in Part 2 of this paper. If this Equation is applied to vertical cylindrical flat-bottom tanks, it becomes (see Appendix of this section)

$$\theta = \frac{D\rho C_p(T - 70)}{4I} + \frac{4.87\, t(T + 20)}{I} \quad \ldots\ldots\ldots [30]$$

in which D is the diameter of the tank in feet. Results calculated by Equation [30] compare favorably with time-temperature observations in Part 1, Table 1.

From Equation [30], it is seen that the time required for the contents to absorb heat at any given rate is principally dependent upon the diameter of the tank. For illustration of the approximate time required to reach 100 F at the observed heat rate, Fig. 19 is shown. The tanks were assumed to be nominally full and of dimensions conforming to many tanks being used at this time. Calculations were made employing properties of compounds having a contents-factor value of 0.136, as shown in Fig. 19. Products having such properties are about as volatile as any stored at atmospheric pressure. Under the assumed conditions, a minimum time element and a maximum venting rate may be expected.

When vessels are in gas or vapor service and exposed to intense fire, the maximum venting rate will be reached in an even shorter time. Fig. 2, Part 1, illustrates the rapid expansion of tank vapors due to fire exposure.

Effect of "Outage." The term "outage" is used in literature pertaining to containers to indicate the proportion of a tank which contains vapor rather than liquid. Thus a tank which is only 40

[33] "American Gas Association Standards," New York, N. Y.

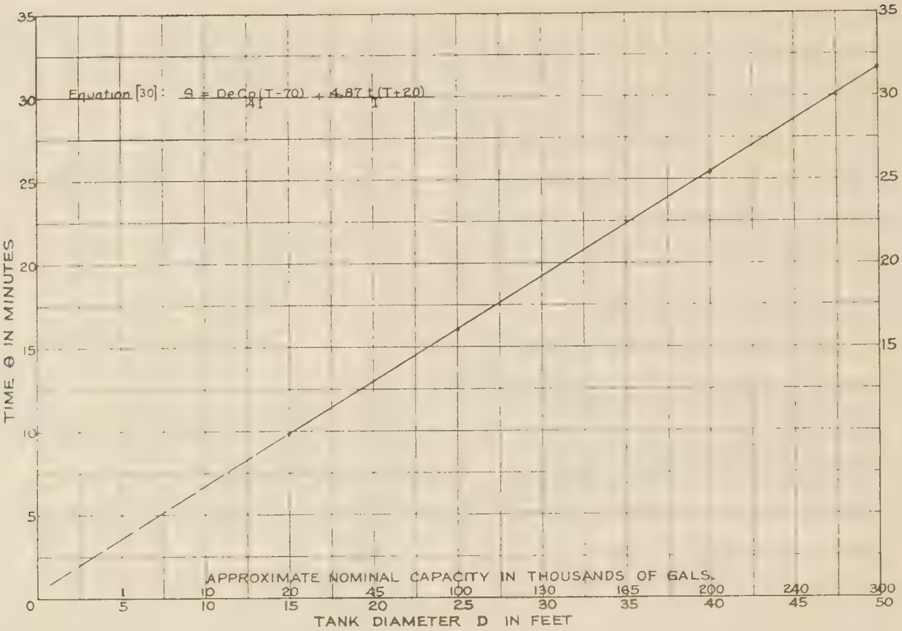


Fig. 19 Estimated Time for Volatile-Liquid Contents of Atmospheric Storage Tanks to Reach 100 F When Absorbing Heat at 20,000 Btu per Sq Ft per Hr

(Products of properties of $F = 0.136$; $T = 104$ F; $M = 74$; $r = 149.5$ Btu; $C_p = 0.447$; $\rho = 54.7$ lb.)

per cent filled would have an "outage" of 60 per cent. Although this term does not appear in the developed formula, it has been considered. If the wetted height of the tank is used rather than the nominally full wetted height, the effect of outage is taken care of. It is true that as the outage increases, the unwetted wall temperature will rise, but heat will be lost by conduction and radiation to the colder portions of the tank, and the temperature of the hot wall will probably not rise above the "critical" point of about 1100 F. When the outage is large, the wetted height is small, and the rate of vaporization is low; and, although the vapor temperature will be higher than the temperature corresponding to the boiling point, it can be shown that the highest venting rate will occur when a tank is full of liquid.

The time required for a completely empty steel tank to reach 1100 F in a fire of intensity I would be approximately

$$\theta = \frac{5114\,t}{0.8\,I} \text{ or } 0.320\,t \text{ for 20,000 Btu input rate} \quad [31]$$

in which θ is time in hours for tank wall to reach 1100 F; t is thickness of tank wall in inches; I is intensity of fire in Btu per hour per square foot. Thus, in a fire of observed intensity of 20,000 Btu per hr per sq ft, a tank made of $^1/_4$-in. steel would reach 1100 F in about 5 min.

$$\theta = 0.320 \times 0.25 = 0.080 \text{ hr} = 4.8 \text{ min}$$

This agrees very closely with time-temperature observations made by the Underwriters' Laboratories, Inc., in their tests described in Part 1. The source of this formula will be found in the Appendix of this section.

Emergency and Normal Relief. The requirements for relief of overpressure due to fire exposure represent an abnormal condition and are greatly in excess of those set forth for normal breathing and displacement. The emergency requirement is adequate for both conditions, and it appears both practical and economical to combine them into one device on the smaller vessels, as tabulated in the Summary for Use.

However, a separate apparatus for the normal requirement has the advantage of greater mechanical dependability over the much larger sizes needed for the combined requirement. Where flash arresters are needed, a further objection arises to large combined units. Efficiently designed arresters, as generally known today, would become huge and expensive, whereas the emergency requirement may be fulfilled separately by merely installing a vaportight hinged cover which may also be used as a manway. (These are available commercially.) If this cover be set to open above 3 in. water pressure, say 2 oz to 4 oz per sq in., no flash arrester would seem needed on this connection. When the emergency valve operated at such pressures, vaporization would already be considerable, and also the vapor would be "too rich to burn," until mixed with external air. Further, flows at 4 oz pressure for compounds having properties of $F = 0.136$ through ordinary vent connections, are of velocities ranging above 25 fps, which is more than the propagation rate of flame in a gas-air mixture at atmospheric pressure.

Modification of the Calculated Relief Areas. The discussion has now progressed so that the suggested reduction of the calculated relief sizes, as shown in Fig. 18, may be clarified.

It will be seen in Fig. 19 that the volatile contents of a 15-ft-diam tank (20,000 gal) may reach 100 F, in a matter of 10 min after being surrounded by intense fire. For this reason, the emergency-vent capacities should be compatible with the 20,000 Btu heat-absorption rate for all tanks from zero through at least 15 ft diam. For tanks above this diameter at these low pressures, the calculated relief areas become rapidly unreasonable (refer to Curve 1, Fig 18), but the likelihood of complete envelopment with fire of sufficient depth (probably about equal to the tank radius) to maintain the heat-input rate, observed in Part 1, becomes less and the time element larger. With these considerations in mind, the suggested vent size (Curve 3, Fig. 18) has been permitted to approach (at 100,000-gal capacity) the curve based on a 6000-Btu heat rate, which curve (the latter) has been gradually flattened to a maximum of 24-in. diam relief size at 300,000 gal capacity. If tanks of this capacity are of 40 to 50 ft diam, as is common, the time element for 100 F is in the neighborhood of

$^1/_2$ hr for severe conditions, which is perhaps sufficient time to employ protective (cooling or extinguishing) equipment reducing the exposure, or abandonment of the vessels.

Conclusion. The conclusions as outlined under *Summary* of this section apply only for the conditions specified.

Equations [20], [21], and [22] are not considered adequate for carbon tetrachloride, are not adequate to vent internal explosion, and are not intended to provide for internal reaction. No reduction in the area of the relief connections is considered for vessels covered with substantial noncombustible insulation, although allowance may be made for this protection in the capacity of the relief devices, which may be changed if the service becomes more severe from the viewpoint of fire exposure, see Proposed Summary for Use.

Equation [23] is not intended to apply to floating-type water-seal gasholders which have, by inherent design, adequate relief for fire exposure.

Suggested sizes of relief connections with necessary capacities for the emergency requirement for atmospheric tanks are given in Table 6 (based on 0.5 psi at 50 gal, proportionally reduced to 3 in. water pressure at 70,000 gal).

TABLE 6 SUGGESTED SIZES OF RELIEF CORRECTIONS FOR EMERGENCY REQUIREMENTS OF ATMOSPHERIC TANKS

Tank capacity, gal	Relief size, in. I.P. size	Minimum capacity, cu ft per hr of air at 60 F, and atmospheric pressure
50	2	13380
50 to 100	3	30100
100 to 300	4	54500
300 to 1000	6	120500
1000 to 2000	8	212000
2000 to 4000	10	326500
4000 to 10000	12	453000
10000 to 20000	16	753000
20000 to 100000	20	1048000
100000 to 200000	22[a]	744000
200000 and up	24	886000

[a] Not a standard pipe size.

Proposed Summary for Use

1 Minimum sizes of relief connections for vessels at pressures not to exceed 1 in. water external, or 0.50 psig internal: All vessels should be provided with relief area (or areas) in the form of connections or hatches. This area should be adequate for both normal breathing and emergency venting generated by exposure to fire.

(*a*) For the normal (based on contraction due to cooling and limited to 1 in. of water external) requirement for vessels in volatile-liquid service use

$$a = \frac{Q}{1172} \text{ or } d = \sqrt{\frac{Q}{921}} \quad \ldots\ldots\ldots\ldots [1]'''$$

in which a is the relief area in square inches, d is the internal diameter of the relief connection in inches, and

$$Q = \frac{S}{0.290} + \text{emptying rate} \quad \ldots\ldots\ldots\ldots [2]'''$$

In Equation [2]''' Q and the emptying rate are in cubic feet of air per hour at 68 F, and atmospheric pressure, and S is the area of the tank roof and shell in square feet.

(*b*) For the emergency requirement (based on 0.50 psi at 50 gal proportionally reduced to 3 in. water internal pressure at 70,000 gal) for vessels not to exceed 15 ft diam in liquid service use

$$a = \frac{0.717\, S_w}{\sqrt{P(P + 14.7)}} \text{ or } d = \sqrt{\frac{0.913\, S_w}{\sqrt{P(P + 14.7)}}} \quad \ldots [3]'''$$

If the pressure limit be 3 in. of water as for atmospheric tanks, Equation [3]''' becomes

$$a = 0.566\, S_w \text{ or } d = \sqrt{0.721\, S_w} \quad \ldots\ldots\ldots\ldots [4]'''$$

and if the atmospheric tank be vertical cylindrical with bottom resting on pad or foundation, Equation [4]''' becomes

$$a = 1.78DH \text{ or } d = \sqrt{2.265\, DH} \quad \ldots\ldots\ldots [5]'''$$

in all of which a is the internal area and d the internal diameter of the relief connection in inches, S_w is the surface in square feet wetted by the contents when nominally full and exposed to fire, P is the relieving pressure in pounds per square inch gage (0.1083 for 3 in. water), D is the diameter of the tank in feet, and H is the height in feet of the contents on the exposed tank wall when nominally full.

For practical reasons, the emergency-vent sizes for all vessels in liquid service above 15 ft diam and, as calculated by Equations [3]''', [4]''', and [5]''' should be modified to conform to a lowered (below 20,000 Btu per sq ft per hr) heat-absorption rate, as proportionally represented by curve 2, Fig. 14, or curve 3, Fig. 18. The ultimate size should be limited to 24 in. diam.

The vent sizes and relief capacities for the normal and emergency requirements (basis as just given) for vertical, cylindrical, above ground (bottom resting on pad), atmospheric storage tanks are given in Table 7.

TABLE 7 VENTING REQUIREMENTS FOR ATMOSPHERIC STORAGE TANKS EXPOSED TO FIRE

	Relief size (I.P. size)		Minimum capacity, cu ft air per hr at 60 F and atmospheric pressure	
Tank capacity, gal	Normal; vent, in.	Emergency; cover, in.	Normal[d] (at 1 in. water flowing pressure)	Emergency (at 3 in. water flowing pressure)
50	...	2[a]	...	6150
50 to 100	...	3[a]	...	13400
100 to 300	...	4[a]	...	24600
300 to 1000	...	6[a]	...	53400
1000 to 2000	1–1$^1/_2$	1–8	1920	98500
2000 to 4000	1–2	1–10	2805	153800
4000 to 10000	1–2	1–12	3660	221500
10000 to 20000	1–3	1–16	6180	394000
20000 to 50000	1–3	1–20	8230	615500
50000 to 100000	1–4	1–20	12500	615500
100000 to 150000	1–4	1–22[e]	14620	744000
150000 to 200000	1–4,[b] 1–3	1–22[e]	18100	744000
200000 to 400000	1–4,[b] 1–3	1–24	22820	886000
400000 to 800000	1–6	1–24	32900	886000
800000 to 1100000	1–6, 1–3	1–24	41100	886000
1100000 to 1300000	1–6,[c] 1–4	1–24	47500	886000
1300000 to 1800000	1–8	1–24	58450	886000
1800000 to 2000000	2–6	1–24	65800	886000

[a] The first four items of "Emergency relief size" are given in terms of "in. vent," all others in terms of "cover, in."
[b] Although one 6 in. may be cheaper, the two units have the advantage of multiple devices.
[c] Although one 8 in. may be cheaper, the two units have the advantage of multiple devices.
[d] To be used for selection of the normal relief devices. This capacity is included in that for the emergency requirement.
[e] Not a standard pipe size.

The size of relief connection as specified in this paragraph should be increased for carbon tetrachloride by multiplying the diameter by 1.14.

(*c*) For the emergency relief requirement for vessels in gas service not to exceed 40 ft diam use

$$a = \frac{0.03195\, S}{\sqrt{P}} \text{ or } d = \sqrt{\frac{0.0407\, S}{\sqrt{P}}} \quad \ldots\ldots\ldots [6]'''$$

in which a is the relief area in square inches, and d is the internal diameter of the relief connection in inches, S is the surface of the tank in square feet which may be exposed including the roof, and P is the relieving pressure in pounds per square inch gage.

Emergency relief connections for vessels in gas service above 40 ft diam should be modified to conform to a lowered heating rate as proportionally represented by curve 5 in Fig. 18, and limited to a maximum of 24 in. diam at 65,000 cu ft capacity.

(*d*) *General.* The relief area should be at least equivalent to

the area of the inlet connection to avoid excessive internal pressure due to accidental overfilling.

In all cases cited, the next larger pipe size should be used as the vent connection, which preferably should be located at the top of the tank vapor space.

No reduction in the size of the relief connection should be made for vessels covered with substantial noncombustible insulation, or provided with reliable cooling systems, although allowance may be made for this protection in the capacity of the relief device. When vessels are so equipped with an adequate relief connection, the relief apparatus may be changed if the service becomes more severe from the standpoint of operation or fire exposure.

2 The relief capacity of the required areas in terms of rate of fluid flow; liquid or gas service: To select a relief device or devices of adequate capacity, based on approved flow test, other than those capacities previously listed for atmospheric tanks, use the area a (not the next larger pipe size), as determined in paragraph 1 in any of the following formulas as the case may require

$$W = 516\,a\sqrt{\frac{MP(P+14.7)}{T}} \text{ or } 405\,d^2\sqrt{\frac{MP(P+14.7)}{T}} \qquad [7]'''$$

lb per hr of vapor contents

$$Q = 198{,}500\,a\sqrt{\frac{P(P+14.7)}{MT}} \text{ or } 156{,}000\,d^2\sqrt{\frac{P(P+14.7)}{MT}} \qquad [8]'''$$

cu ft per hr of vapor at 68 F, and atmospheric pressure

$$Q = 36{,}900\,a\sqrt{\frac{P(P+14.7)}{T}} \text{ or } 29{,}000\,d^2\sqrt{\frac{P(P+14.7)}{T}} \qquad [9]'''$$

cu ft per hr of air at 68 F, and atmospheric pressure

In all of which a is the internal area in square inches, d is the diameter of the relief area in inches, P is the relieving pressure in pounds per square inch (not to exceed 0.50 lb), M is the molecular weight. and T is the vapor temperature of the contents in deg F absolute.

For conditions of fire exposure, T is approximately the normal boiling point of the liquid contents. For vessels in gas service, T is about the temperature necessary to raise the pressure in the vessel from the absolute working pressure and temperature to the absolute set pressure of the relief device, as computed by the gas law, $T/P = T'/P'$.

Allowance may be made in the capacity of the relief device for vessels covered with substantial noncombustible insulation or provided with reliable cooling systems proportional to the reduction effected in the heat-transfer rate through the tank wall, if it is deemed advisable from a practical and economical standpoint.

3 Normal and emergency relief devices: It is preferable, as explained in the section, *Emergency and Normal Relief* that separate relief devices, when needed, be used for the venting requirement for normal operation for all vessels above 1000 gal capacity. The additional relief area required for fire exposure may be in the form of a vaportight hinged cover, hatch, or simple valve, each of which is considered more practical than a weak head seam.

Uncontrolled internal reaction and explosion in vessels of the subject category are emergency relief requirements to which this development does not apply. However, weak seams have been advantageous under such circumstances. The hazard of internal explosion may be minimized by the use of inert gas, grounding the vessel, or provision of flash arresters.

4 If relief capacity is inadequate for the contingency of fire exposure, additional protective features should be provided for the vessel and exposures. These may consist of fire-protective or extinguishment equipment, process-cooling systems, construction of dikes or walls, provision of drains, or fireproofing supports, etc.

5 Back pressure: In this low-pressure range, any back pressure on the downstream side of the relief device will reduce its flow capacity, as calculated by Equations [7]''', [8]''', and [9]''' proportional to the $\sqrt{P(P+14.7) - P'(P'+14.7)/P(P+14.7)}$ in which P' is the back pressure. Also, the opening pressure of the relief device must be adjusted to counteract the back pressure or the intended maximum pressure in the vessel will be exceeded.

6 Time element: The time required for the liquid contents of a vessel when exposed to fire to reach the boiling point (or any given average temperature) may be estimated by the following formula

$$\theta = \frac{V\rho C_p(T-70)}{IS_w} + \frac{4.87t(T+20)}{I} \qquad [10]'''$$

in which θ is the time in hours, V is the volume of the contents in cubic feet, ρ is the density of contents in pounds per cubic foot, C_p is the specific heat of contents, T is the temperature reached in deg F, I is the heat-absorption rate in Btu per hour per square foot, and t is the thickness of the steel tank wall in inches. If Equation [10]''' is to apply to vertical, cylindrical, flat-bottom tanks, it becomes

$$\theta = \frac{D\rho C_p(T-70)}{4I} + \frac{4.87\,t(T+20)}{I} \qquad [11]'''$$

in which D is the diameter of the tank in feet.

If θ is $^1/_2$ hr or more (approximate time for very volatile materials to reach 100 F, when contained in 40 to 50-ft-diam atmospheric tanks surrounded by intense fire) and there is assurance that effective cooling or fire-extinguishing equipment will come into play during the $^1/_2$-hr period, the emergency-relief (device) capacity may be reduced to the vaporization rate produced by the calculated temperature rise within the $^1/_2$ hr. For vessels exposed to fire in gas service, the maximum venting rate will occur within such a short time as to permit no modification of the calculated emergency relief.

Conclusion

The conclusions as outlined under *Summary* in the first of this section of the paper and as detailed under Summary for Use are based upon the foregoing theory which is mathematically expressed in the Appendix of this section.

The formulas proposed provide relief capacities greater than those provided by present average practice which capacities are not adequate for the abnormal condition of fire exposure.

For examples of use of the formulas see the Appendix.

Proposals herein presented hold only for pressures not to exceed 0.50 psig. For vessels to operate at pressures from 0.50 to include 15 psig, Part 4 of this paper will be presented at a later date. For pressures above that refer to Part 2.

Appendix for Part 3

Derivation of General Equations [24] or [3]^{iv}, and [25] or [15]^{iv}

1 Vapor weight rate of a short tube (at pressures not to exceed 0.50 psig), Equation [26]

$$V = C\sqrt{2g\Delta H}$$

$$Va = Ca\sqrt{2g\Delta H}$$

$$Va\rho = Ca\rho\sqrt{2g\Delta H}$$

$$3600Va\rho = 3600Ca\rho\sqrt{2g\Delta H}$$

$$W = 3600Ca\rho\sqrt{2g\Delta H}$$

When differential is pounds per square inch

$$\Delta H = \frac{\Delta P\ 144}{\rho^*}$$

$$\Delta H = \frac{\Delta P \times 10.72 \times T \times 144}{MP^*}$$

$$W = 3600 \times 0.70 \times \frac{0.7854d^2}{144} \times \frac{MP}{10.72T}\sqrt{\frac{64.35 \times \Delta P \times 10.72 \times T \times 144}{MP^*}}$$

$$W = 405\frac{d^2\,MP}{T}\sqrt{\frac{\Delta PT}{MP^*}}$$

$$W = 405\,d^2\sqrt{\frac{\Delta PMP}{T}} \ldots\ldots\ldots\ldots\ [1]^{\text{iv}}$$

In the development of Equation $[1]^{\text{iv}}$ the following nomenclature was used:

- V = velocity, fps
- g = gravitational constant, 32.17
- ΔH = differential head in ft of fluid*
- C = coefficient of discharge
- C = 0.70 when $p_1/p_2 < 1.05$ (see footnote 31)
- a = area of short tube, sq ft
- ρ = density of fluid, lb per cu ft
- ρ = MP/RT for a gas
- R = gas constant, 10.72
- W = weight rate, lb per hr
- M = molecular weight
- P = absolute pressure, psi
- T = absolute temperature, deg F
- ΔP = differential pressure, psi
- d = diameter of a short tube, in.

2 Weight rate vaporized

$$\frac{Q}{S_w} = \frac{Wr}{S_w} = I$$

$$W = \frac{IS_w}{r} \ldots\ldots\ldots\ldots\ [2]^{\text{iv}}$$

in which

- Q = total heat input, Btu per hr
- S_w = wetted surface heated, sq ft
- r = latent heat of vaporization, Btu per lb
- I = unit heat input, Btu per hr per sq ft
- W = liquid vaporized, lb per hr

* It is theoretically correct to express the differential head ΔH in terms of the downstream density and so the P in the denominator in downstream pressure. Cancelling this P into the upstream pressure simplifies the equation, avoids confusion, and yet maintains theoretical accuracy within 0.36 per cent at 3 in. water pressure (1.7 per cent at 0.50 lb per sq in. gage). Note the comparison between calculated and actual flows in Fig. 20.

3 Required diameter of a short tube to pass the weight rate vaporized, Equation [24] or $[3]^{\text{iv}}$:

Equate $[1]^{\text{iv}}$ to $[2]^{\text{iv}}$

$$405\,d^2\sqrt{\frac{\Delta PMP}{T}} = \frac{IS_w}{r}$$

$$d^2 = \frac{IS_w}{r \times 405}\sqrt{\frac{T}{M\Delta PP}}$$

$$d = \sqrt{\frac{IS_w}{405\sqrt{\Delta PP}}\left(\frac{T}{Mr^2}\right)^{1/4}} \ldots\ [3]^{\text{iv}} \text{ or } [24]$$

To apply Equation $[3]^{\text{iv}}$

- d = diameter of relief connection, in.
- I = 20,000 Btu per sq ft per hr
- S_w = wetted surface exposed, sq ft
- ΔP = relieving pressure, psig
- P = relieving pressure, psia
- T = boiling point of contents at P, deg F abs
- M = molecular weight of contents
- r = latent heat of vaporization at T, Btu per lb

4 Weight rate of expansion or contraction of gas or vapor due to heat transferred to or from the tank wall:

$$wC_p\frac{dT}{d\theta} = \frac{Q}{\theta} \ldots\ldots\ldots\ldots\ [4]^{\text{iv}}$$

$$hS\Delta T = \frac{Q}{\theta} \ldots\ldots\ldots\ldots\ [5]^{\text{iv}}$$

$$w = V\rho = V\frac{MP}{RT} \ldots\ldots\ldots\ldots\ [6]^{\text{iv}}$$

From Equations $[4]^{\text{iv}}$ and $[6]^{\text{iv}}$

$$V\rho C_p\frac{dT}{d\theta} = \frac{Q}{\theta} \ldots\ldots\ldots\ldots\ [7]^{\text{iv}}$$

$$h = F\overline{\Delta T}^{1/4} \text{ (see Part 1, ref 5)}$$

From Equation $[5]^{\text{iv}}$

$$F\overline{\Delta T}^{1/4}\,S\Delta T = \frac{Q}{\theta} \ldots\ldots\ldots\ldots\ [8]^{\text{iv}}$$

Equate $[7]^{\text{iv}}$ to $[8]^{\text{iv}}$

$$V\rho C_p\frac{dT}{d\theta} = FS\overline{\Delta T}^{5/4}$$

$$\frac{dT}{d\theta} = \frac{FS\overline{\Delta T}^{5/4}}{V\rho C_p} \ldots\ldots\ldots\ldots\ [9]^{\text{iv}}$$

From Equation $[6]^{\text{iv}}$

$$dw = \frac{-VMP}{R} \cdot \frac{dT}{T^2} \ldots\ldots\ldots\ldots\ [10]^{\text{iv}}$$

$$-dw = \frac{w}{\theta} \cdot d\theta \ldots\ldots\ldots\ldots\ [11]^{\text{iv}}$$

Equate $[10]^{\text{iv}}$ to $[11]^{\text{iv}}$

$$\frac{VMP}{R} \cdot \frac{dT}{T^2} = \frac{w}{\theta} \cdot d\theta \ldots\ldots\ldots\ldots\ [12]^{\text{iv}}$$

$$\frac{dT}{d\theta} = \frac{w}{\theta} \cdot \frac{RT^2}{VMP} \ldots\ldots\ldots\ldots\ [13]^{\text{iv}}$$

Equate [9]^iv to [13]^iv

$$\frac{FS\overline{\Delta T}^{5/4}}{V\rho C_p} = \frac{w}{\theta} \cdot \frac{RT^2}{VMP} \text{.............} [14]^{iv}$$

In the development of Equations [4]^iv to [14],^iv the following nomenclature was used:

- w = weight of contained gas or vapor, lb
- C_p = specific heat of gas or vapor
- dT = change in temperature
- $d\theta$ = change in time
- Q = total heat transferred, Btu
- θ = time, unit hr
- h = heat-transfer coefficient Btu per hr per sq ft per deg F
- S = tank surface exposed, sq ft
- ΔT = difference between tank wall and contained gas, deg F
- V = volume of contents gas or vapor, cu ft
- ρ = density of contents gas, lb per cu ft
- M = molecular weight of gas or vapor
- P = pressure, psia
- R = gas constant, 10.72
- T = temperature of gas or vapor, deg F abs
- F = function of physical properties of gas
- dw = change in weight
- $W = \frac{w}{\theta}$ = weight rate, lb per hr

5 Required diameter of a short tube to pass the weight rate of a gas expanded or contracted due to heat transferred to or from the tank wall; Equation [25] or [15]^iv (same nomenclature used as in section 4 preceding; also $\rho = MP/RT$):

Insert Equation [1]^iv in [14]^iv

$$405\, d^2 \sqrt{\frac{\Delta PMP}{T}} \cdot \frac{RT^2}{VMP} = \frac{FS\overline{\Delta T}^{5/4}}{V\rho C_p}$$

$$d^2 = \frac{FS\overline{\Delta T}^{5/4}}{V\rho C_p} \cdot \frac{VMP}{RT^2\, 405} \sqrt{\frac{T}{\Delta PMP}}$$

$$d^2 = \frac{F}{C_p M^{1/2}} \cdot \frac{S}{405} \cdot \frac{\overline{\Delta T}^{5/4}}{T} \sqrt{\frac{T}{\Delta PP}}$$

$$d = \left(\frac{F}{C_p M^{1/2}}\right)^{1/2} \cdot \frac{S^{1/2}}{\sqrt{405}} \cdot \frac{\overline{\Delta T}^{5/8}}{T^{1/4}} \cdot \frac{1}{\Delta P^{1/4} P^{1/4}}$$

$$d = \frac{0.375 P^{1/4}}{T^{0.753}} \cdot \frac{S^{1/2}}{\sqrt{405}} \cdot \frac{\overline{\Delta T}^{5/8}}{T^{1/4}} \cdot \frac{1}{\Delta P^{1/4} P^{1/4}}$$

(Refer to the following paragraph for $\frac{0.375\, P^{1/4}}{T^{0.0753}}$)

$$d = 0.0186 \cdot \frac{S^{1/2}}{\Delta P^{1/4}} \cdot \frac{(T_w - T)^{5/8}}{T^{0.325}} \text{.............} [15]^{iv} \text{ or } [25]$$

To apply equation [15]^iv

- d = diameter of relief connection, in.
- S = tank surface heated, sq ft
- ΔP = relieving pressure, psig
- T_w = temperature to which tank wall is heated or cooled, deg F abs
- T = initial temperature of contents vapor, deg F abs; if T_w is less than T, reverse the signs

The term $\left(\frac{F}{C_p M^{1/2}}\right)^{1/2}$ in this section, relating the properties of the vapor contents, was replaced by its maximum value $0.375 \cdot \frac{P^{1/4}}{T^{0.0753}}$. This is justified in the following manner: The value of F, a function of the physical properties of a gas, varies with both temperature and pressure as will be seen. The value C_p specific heat at constant pressure varies with temperature. Inasmuch as pressure and temperature are working conditions to remain in the relief diameter Equation [15]^iv, it is desirable to express the subject variable in terms of P and T. Now

$$F = D \left(\frac{\rho^2 B C_p k^3}{Z}\right)^{1/4} \quad \text{(Part 1, ref. 5, Equation [20], p. 242)}$$

$$\left(\frac{F}{C_p M^{1/2}}\right)^{1/2} = D \left(\frac{\rho^2 B C_p k^3}{C_p^4 M^2 Z}\right)^{1/8} = D \left[\rho^2 Z^2 \cdot \frac{B}{M^2} \left(\frac{k^3}{C_p^3 Z^3}\right)\right]^{1/8}$$

Since B, M, and the ratio $C_p Z/k$ (Part 1, ref. 5, p. 417) are independent of P and T, we may write

$$\left(\frac{F}{C_p M^{1/2}}\right)^{1/2} = D' (\rho^2 Z^2)^{1/8}$$

and ρ for a gas is MP/RT, then

$$\left(\frac{F}{C_p M^{1/2}}\right)^{1/2} = D' \left(\frac{M^2 P^2}{R^2 T^2} \cdot Z^2\right)^{1/8}$$

Again M and R are independent of P and T, so

$$\left(\frac{F}{C_p M^{1/2}}\right)^{1/2} = D'' \left(\frac{P^2}{T^2} \cdot Z^2\right)^{1/8}$$

$$= D'' \left(\frac{P^2}{T^2} \cdot T^{1.398}\right)^{1/8}$$

$$= D'' \left(\frac{P^2}{T^{0.602}}\right)^{1/8}$$

$$\left(\frac{F}{C_p M^{1/2}}\right)^{1/2} = D'' \left(\frac{P^{1/4}}{T^{0.0753}}\right)$$

When $P = 14.7$ psia, and $T = 528$ deg R, $C_p = 0.24$, and $F = 0.27$ for air

$$\therefore D'' = \left(\frac{0.27}{0.24 \times 29^{1/2}}\right)^{1/2} \times \frac{(528)^{0.0753}}{14.7^{1/4}}$$

$$D'' = 0.375 \text{ and } \quad \left(\frac{F}{C_p M^{1/2}}\right)^{1/2} = 0.375\, \frac{P^{1/4}}{T^{0.0753}}$$

The nomenclature used in the foregoing derivations follows:

- F = 0.27 for air at 68 F, and atmospheric pressure
- D = proportionality constant
- k = thermal conductivity
- C_p = specific heat
- ρ = density
- B = coefficient of thermal expansion
- Z = viscosity
- M = molecular weight
- R = gas constant

Z^2 varies with T for air as follows:

T, deg R	Z	Z^2
460	0.0162	0.0002625
528	0.018	0.000323
660	0.021	0.000440
860	0.025	0.000625
1060	0.029	0.000840
1260	0.032	0.001050
1460	0.036	0.001295
1560	0.038	0.001444

$$\left(\frac{1560}{460}\right)^n = (3.39)^n = \frac{1.444}{0.2625} = 5.50$$

$$n = 1.398$$

TABLE 8 VALUES OF F FOR SEVERAL GASES

Gas	k^a	$C_p{}^a$	Z^a	M^a	$\frac{k^3C_pM^2}{Z}$	$\left(\frac{k^3C_pM^2}{Z}\right)^{1/4}$	F	$\frac{F}{C_pM^{1/2}}$
Air	0.0161	0.24	0.018	29	0.0468	0.465	0.270	0.209
Hydrogen	0.1075	3.50	0.009	2	1.9324	1.181	0.685	0.138
Acetone	0.0065	0.34	0.0077	58	0.0408	0.448	0.260	0.100
Acetylene	0.0124	0.43	0.010	26	0.0554	0.485	0.281	0.128
Benzene	0.0062	0.26	0.0075	78	0.0502	0.472	0.273	0.135
Ethyl alcohol	0.0089	0.40	0.0094	46	0.0634	0.501	0.290	0.107
Ethyl ether	0.0088	0.46	0.0077	74	0.2220	0.687	0.398	0.100
Water	0.0110	0.48	0.0097	18	0.0213	0.382	0.221	0.108
Pentane	0.0083	0.34	0.0063	72	0.1599	0.632	0.366	0.127
Carbon dioxide	0.0109	0.21	0.0146	44	0.0360	0.436	0.252	0.181
"Freon 113"	0.0054	0.16	0.0103	187	0.0855	0.542	0.313	0.143

[a] "Heat Transmission," by W. H. McAdams,[3] various pages.

Now it is also necessary to show that $\left(\frac{F}{C_pM^{1/2}}\right)^{1/2}$ gives higher values for air than for any other gases whose properties were available in the reference literature. To do this, we use McAdam's Equation [20][34] again which is

$$F = D\left(\frac{\rho^2BC_pk^3}{Z}\right)^{1/4}$$

and in which the nomenclature is as before. Again, $\rho = MP/RT$ and $B = 1/492$. Then, at a constant pressure and temperature it would be expected that F would vary for different gases as follows.

$$F = D'\left(\frac{M^2C_pk^3}{Z}\right)^{1/4}$$

At 1 atm pressure and 20 C, $F = 0.27$ and $\left(\frac{M^2C_pk^3}{Z}\right)^{1/4} = 0.465$ for air, so that the F value for any other gas, whose molecular weight, specific heat, thermal conductivity, and viscosity are known, may be determined from the relation.

$$F_g = \frac{0.27}{0.465}\left(\frac{k^3C_pM^2}{Z}\right)_g^{1/4}$$

in which the subscript g applies to the gas other than air. On this basis the F value for a number of representative gases has been calculated as shown in Table 8. It will be noted that the value of $\left(\frac{F}{C_pM^{1/2}}\right)^{1/2}$ for air is greater than for any of those so obtained, and for many others such as ammonia, nitrogen, and oxygen whose properties were considered but not listed in the table.

Simplification of General Expressions to Formulas For Use

1 The diameter of a short tube to limit internal pressure (not to exceed 0.50 psig) within vessels when the volatile-liquid contents are absorbing heat at 20,000 Btu per sq ft per hr of wetted surface exposed, Equations [20], [21], and [22], is derived as follows, the nomenclature being the same as used previously

$$d = \sqrt{\frac{IS_w}{405\sqrt{\Delta PP}}}\left(\frac{T}{Mr^2}\right)^{1/4} \ldots\ldots [3]^{\mathrm{iv}} \text{ or } [24]$$

in which $\left(\frac{T}{Mr^2}\right)^{1/4}$ = contents factor, evaluated from Fig. 12, Part 2.

$$d = \sqrt{\frac{20{,}000\,S_w}{405\sqrt{P(P+14.7)}}} \times 0.136$$

$$d = \sqrt{\frac{0.913S_w}{\sqrt{P(P+14.7)}}} \ldots\ldots [16]^{\mathrm{iv}}$$

If P be 3 in. water, 0.1083 lb

$$d = \sqrt{\frac{0.913S_w}{\sqrt{0.1083 \times 14.8083}}} = \sqrt{0.721S_w} \ldots\ldots [17]^{\mathrm{iv}}$$

If the tank be vertical cylindrical with bottom on pad

$$d = \sqrt{0.721\pi DH} = \sqrt{2.265DH} \ldots\ldots [18]^{\mathrm{iv}}$$

2 The diameter of a short tube to limit internal pressure to a specified maximum (not to exceed 0.50 psig) within vessels when the vapor contents absorb heat from the enclosing shell heated to 1100 F, Equation [23], is derived as follows, the nomenclature being the same as previously used:

$$d = 0.0186\,\frac{S^{1/2}}{\Delta P^{1/4}} \cdot \frac{(T_w - T)^{5/8}}{T^{0.325}} \ldots [15]^{\mathrm{iv}} \text{ or } [25]$$

in which $\frac{(T_w - T)^{5/8}}{T^{0.325}}$ = gas expansion variable. Then substituting the following values:

$$T_w = 1100 + 460 = 1560 \text{ deg F abs}$$

$$T = 0 + 460 = 460 \text{ deg F abs}$$

$$(T_w - T)^{5/8} = 79.5$$

$$T^{0.325} = 7.33$$

$$(T_w - T)^{5/8}/T^{0.325} = 10.85$$

$$d = 0.0186 \times \frac{S^{1/2}}{\Delta P^{1/4}} \times 10.85$$

$$d = \sqrt{\frac{0.0407\,S}{\sqrt{P}}} \ldots\ldots [19]^{\mathrm{iv}}$$

when P is gage pressure.

3 The diameter of a short tube to limit external pressure on a tank to 1 in. of water when the vapor contents are contracted from the loss of heat to a colder tank wall, Equations [16], [17], [18], and [19], is derived as follows, the nomenclature being the same as previously used.

The weight rate of contraction is

$$W = \frac{FS\Delta T^{5/4}}{V\rho C_p}\;\frac{VMP}{RT^2} \ldots\ldots [14]^{\mathrm{iv}}$$

$$Q = \frac{FS\Delta T^{5/4}}{C_p} \cdot \frac{MP}{RT^2} \cdot \frac{R^2T^2}{M^2P^2}$$

$$Q = \frac{10.72\,FS(T - T_w)^{5/4}}{C_pMP} \text{ cu ft per hr}$$

in which

[34] Part 1, ref. 5, equation [20], p. 242.

$$\frac{\text{Weight rate, } W}{\text{Unit weight, } \rho} = \text{quantity rate, } Q$$

$$\rho = MP/RT$$

Applying the temperatures from Case 1, Table 4, and the properties of air

$$Q = \frac{10.72 \times 0.27 \times S(563 - 516)^{5/4}}{0.24 \times 29 \times 14.736}$$

(14.736 = average pressure during contraction)

$$Q = S/0.291 \text{ cu ft per hr} \ldots\ldots\ldots\ldots [20]^{\text{iv}}$$

When Equation $[20]^{\text{iv}}$ is used, the emptying rate must be added to provide for maximum in-breathing.

The weight rate of a relief connection (nomenclature as previously used) is

$$W = 405\, d^2 \sqrt{\frac{\Delta PMP}{T}} \ldots\ldots\ldots\ldots [1]^{\text{iv}}$$

$$Q = 405\, d^2 \sqrt{\frac{\Delta PMP}{T}} \cdot \frac{RT}{MP}$$

again $Q = \dfrac{W}{\rho}$

$$Q = 4340\, d^2 \sqrt{\frac{\Delta PT}{MP}} \text{ cu ft per hr} \ldots\ldots\ldots [21]^{\text{iv}}$$

(see Fig. 20).

Then the quantity rate of air at 68 F is

$$Q = 4340\, d^2 \sqrt{\frac{0.0361 \times 528}{29 \times 14.736^*}} = 918\, d^2 \text{ at 1 in. water pressure}$$

$$d = \sqrt{\frac{Q \times 14.7}{918 \times 14.736}} = \sqrt{Q/921} \text{ at atmospheric pressure } [22]^{\text{iv}}$$

Applying the surface from Case 1, Table 4, to Equations $[20]^{\text{iv}}$ and $[22]^{\text{iv}}$

$$Q = S/0.290 = 5170/0.290 = 17{,}830 \text{ cu ft of air}$$

$$d = \sqrt{Q/921} = \sqrt{17{,}830/921} = 4.40 \text{ in.}$$

Check by applying Equation $[15]^{\text{iv}}$ to Case 1, Table 4

$$d = 0.0186 \times \frac{S^{1/2}}{\Delta P^{1/4}} \times \frac{(T - T_w)^{5/8}}{T^{0.325}} \ldots\ldots [15]^{\text{iv}}$$

the following values being used:

$$S^{1/2} = \sqrt{5170} = 71.9$$

$$\Delta P^{1/4} = \sqrt[4]{0.0361} = 0.436$$

$$T^{0.325} = (563)^{0.325} = 7.82$$

$$T_w = 516 \text{ F abs}$$

$$(T - T_w)^{5/8} = (47)^{5/8} = 11.1$$

$$(T - T_w)^{5/8}/T^{0.325} = 1.425$$

then

$$d = \frac{0.0186 \times 71.9 \times 1.425}{0.436}$$

$$d = 4.37 \text{ in.}$$

A discrepancy exists of 3 parts in 438.5 between results by Equations $[15]^{\text{iv}}$ and $[22]^{\text{iv}}$. This is considered due to the average temperature difference between the tank vapor and external air during in-breathing for which Equation $[15]^{\text{iv}}$ does not provide.

4 The capacity of the relief areas in terms of fluid flow, Equations [26], [27], [28], [7]‴, [8]‴, and [9]‴.

The weight rate of vapor flow of a short tube is

$$W = 405\, d^2 \sqrt{\frac{MP(P + 14.7)}{T}} \text{ lb per hr} \ldots\ldots\ldots [1]^{\text{iv}}$$

The quantity rate is

$$Q = \frac{W \times 10.72 \times 528}{M \times 14.7}$$

where

$$Q = W/\rho$$

$$\rho = MP/RT$$

$$\rho = M\,(14.7)/10.72\,(528)$$

$$Q = 156{,}000\, d^2 \sqrt{\frac{P(P + 14.7)}{MT}} \ldots\ldots\ldots [23]^{\text{iv}}$$

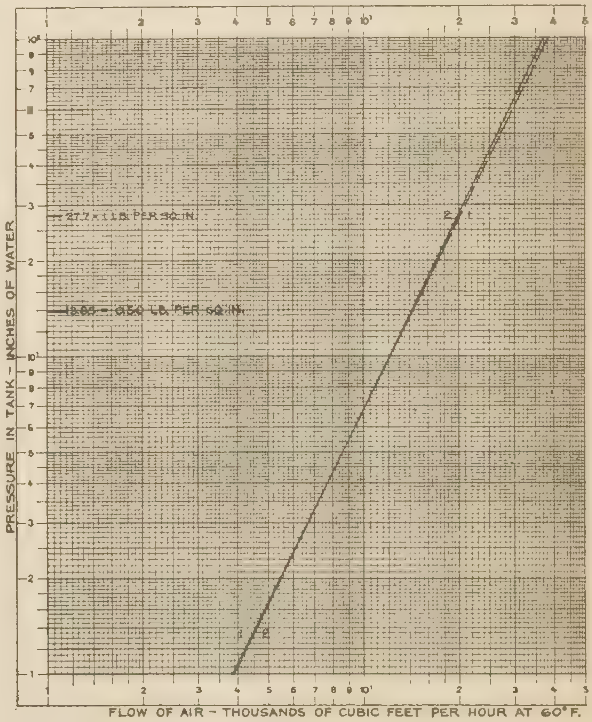


Fig. 20 Comparison of Theoretical Flow Capacity of Short Tube With That of Typical Tank Connection at Low Pressures

Notes: 1 Actual flows through a calibrated orifice to the tank.

2 Theoretical flow capacity equation $Q = 4340\, d^2 \sqrt{\dfrac{\Delta PT}{MP}}$ (coefficient of flow 0.70).

Refer to Part 3, "Relief of Vessels Exposed to Fire."

Capacity of a standard 2-in. nipple 6 in. long located in top head of 4 ft diameter by 5 ft high vertical cylindrical tank with inlet in lower shell. Actual flows were 99 per cent of theoretical at 1 in. water pressure.

* Actually, the external pressure during contraction cannot rise above 14.7 lb per sq in. absolute. Flows computed using 14.736 as P are within the accuracy illustrated by Fig. 20. Use of the P (14.736) is more than compensated for by using $Q = S/0.290$ instead of $S/0.291$.

cu ft per hr at 68 F and atmospheric pressure

The quantity rate of air is

$$Q = 156{,}000\, d^2 \sqrt{\frac{P(P + 14.7)}{29\,T}}$$

$$Q = 29{,}000\, d^2 \sqrt{\frac{P(P + 14.7)}{T}} \quad\ldots\ldots\ldots\ [24]^{\text{iv}}$$

cu ft per hr at 68 F and atmospheric pressure

At a flowing temperature of 104 F such as for compounds of $F = 0.136$

$$Q = 29{,}000\, d^2 \sqrt{\frac{P(P + 14.7)}{564}}$$

$$Q = 1222\, d^2 \sqrt{P(P + 14.7)} \quad\ldots\ldots\ldots\ [25]^{\text{iv}}$$

cu ft air per hr at 68 F and atmospheric pressure

When $P = 0.1083$ lb, 3 in. water

$$Q = 1550\, d^2 = 1972a \quad\ldots\ldots\ldots\ [26]^{\text{iv}}$$

cu ft air per hr at 68 F and atmospheric pressure

Derivation of Time-Element Formulas

1 Estimated time for the liquid contents of a vessel to reach a given temperature, Equations [29] and [30]

IS_w = total heat input, Btu per hr

$\frac{V\rho C_p\Delta T}{\theta}$ = heat absorbed by liquid contents, Btu per unit time

$\frac{St\rho C_p\Delta T}{12\,\theta}$ = heat absorbed by tank, Btu per unit time

$$IS_w = \frac{V\rho C_p(T - 70)}{\theta} + \frac{St\,487 \times 0.12\,(T + 20)}{12\,\theta}$$

$$(S_w \leq 0.85\,S)$$

$$\theta = \frac{V\rho C_p\,(T - 70)}{IS_w} + \frac{4.87\,t(T + 20)}{I} \quad\ldots\ [27]^{\text{iv}}$$

(Liquid contents of an average temperature of 70 F and tank steel of an average of 60 F before exposure. Tank shell after heating of an average of 100 F hotter than liquid, although this will vary with the contents.)

If tank is vertical cylindrical with bottom on pad, $S_w = \pi DH$

$$\theta = \frac{D\rho C_p(T - 70)}{4I} + \frac{4.87\,t\,(T + 20)}{I} \quad\ldots\ldots\ [28]^{\text{iv}}$$

in which

- I = heat absorption rate, Btu per sq ft per hr
- S_w = surface wetted by contents exposed to fire, sq ft
- V = volume of liquid contents, cu ft
- ρ = density, lb per cu ft (487 for steel)
- C_p = specific heat, 0.12 for steel
- ΔT = temperature difference, deg F
- θ = time, hr
- S = surface of vessel heated, sq ft
- t = thickness of tank shell in inches
- T = temperature of contents, deg F
- D = diameter of vessel, ft
- H = height of contents on exposed wall, ft

2 Estimated time for an empty steel tank to reach the "critical" temperature of 1100 F, Equation [31], the foregoing nomenclature being used

$$IS = \frac{St}{12} \times \frac{487 \times 0.12\,(T - 50)}{\theta}$$

$$I = \frac{4.87\,t(1100 - 50)}{\theta}$$

$$\theta = \frac{5114\,t}{0.8\,I} = 0.320\,t \quad\ldots\ldots\ldots\ [29]^{\text{iv}}$$

(During the period of heating to 1100 F, the input I will be only about 80 per cent effective due principally to reradiation losses.)

Examples of Use of the Formulas

1 Assume an acetone tank (110,000 gal nominal capacity) 30 ft diam by 24 ft high, cylindrical shell with umbrella-type roof and flat bottom. The relief devices are to operate at 1 in. water external and 0.50 psi internal pressure.

(a) The normal breathing requirement (based on in-breathing) was determined as explained and tabulated in the Summary for Use in this section of the paper. This was done as follows

$$Q = (S/0.290) + \text{emptying rate} \quad\ldots\ldots\ldots\ [2]'''$$

$$Q = (3013/0.290) + 300 \text{ gal per min}$$
$$Q = 10{,}400 + 2408 \text{ cu ft per hr}$$
$$Q = 12{,}808 \text{ cu ft air per hr at 68 F}$$
$$Q = 12{,}808 \sqrt{\frac{520}{528}} = 12{,}710 \text{ at 60 F}$$
$$d = \sqrt{Q/921} \quad\ldots\ldots\ldots\ [1]'''$$
$$d = \sqrt{12{,}808/921} = 3.73 \text{ in. (specify 4-in. standard I.P. size)}$$

In this

$$S = \pi\,D(H + 0.266D)$$
$$H = \text{height of straight shell}$$
$$S = \pi\,30\,(24 + 0.266 \times 30)$$
$$S = 3013 \text{ sq ft}$$

The 4-in. size indicated is that recommended for this tank since it may be used at atmospheric pressure. However, the requirement may be figured more exactly (provide for the difference between 3 in. water and 0.50 psi internal pressure) as follows:

$$Q = \frac{S}{0.291 \times \frac{14.932}{14.736}} = S/0.295$$

See Equation [20]$^{\text{iv}}$

$$Q = (3013/0.295) + 2408$$
$$Q = 12{,}623 \text{ cu ft air per hr at 68 F}$$
$$d = \sqrt{Q/921} \quad\ldots\ldots\ldots\ [1]'''$$
$$d = \sqrt{12{,}623/921} = 3.70 \text{ in. (use 4 in. as before)}$$

(b) The emergency vent size at 3 in. water pressure is also tabulated in the Summary for Use of this section of this paper and is that recommended. Considerations involved in its selection are as follows

$$d = \sqrt{\frac{0.913S_w}{\sqrt{P(P + 14.7)}}} \quad\ldots\ldots\ldots\ [3]'''$$

$$d = \sqrt{\frac{0.913 \times 2260}{\sqrt{0.50(15.2)}}} = 27.37 \text{ in.}$$

where

$$S = \pi\,DH$$
$$S = \pi \times 30 \times 24$$
$$S = 2260 \text{ sq ft}$$

This is seen to be larger than the 24 in. maximum diam recommended (based on the lengthened time element and lessened pos-

sibilities of completely effective total heat absorption) for any vessel. It incorporates the observed 20,000 Btu per sq ft per hr heat-absorption rate.

The size (22 in.) recommended was based on a modified rate at 3 in. water pressure which rate may be obtained as follows

$$d = \sqrt{\frac{IS_w}{405\sqrt{P(P+14.7)}}}\left(\frac{T}{Mr^2}\right)^{1/4} \ldots\ldots [24]$$

$$\sqrt{484} = \sqrt{\frac{I \times 2260}{405\sqrt{0.1083(14.808)}}} \times 0.136$$

$$I = \frac{484 \times 405 \times 1.268}{2260 \times 0.0185} = 5{,}950 \text{ Btu}$$

On the same modified rate basis, d for 0.50 psi relief pressure would become

$$d = 27.37\sqrt{\frac{5950}{20{,}000}} = 14.92 \text{ in.}$$

If this is used and the tank becomes involved with fire, the possibilities of rupture are greater.

If the recommended 22 in. be provided, the designed heat-absorption rate at 0.50-psi relief pressure is

$$I = 5950\sqrt{\frac{0.50(15.2)}{0.1083(14.808)}} = 12{,}950 \text{ Btu per sq ft per hr}$$

The relief capacity of the 22-in. connection at 3 in. water flowing pressure is

$$Q = 1222\, d^2 \sqrt{P(P+14.7)} \ldots\ldots [25]^{\text{iv}}$$

$$Q = 1222 \times 484\sqrt{0.1083 \times 14.808}$$

Q = 750,000 cu ft of free air per hr at 68 F (based on 104 F flowing temperature)

Compared with acetone vapor

$$Q = 4340\, d^2\sqrt{\frac{\Delta PT}{MP}} \ldots\ldots [19]$$

$$Q = 4340 \times 484\sqrt{\frac{0.1083 \times 593}{58 \times 14.808}}$$

Q = 575,000 cu ft of acetone per hr at 3 in. water and 133 F

$$Q = 575{,}000\sqrt{\frac{528 \times 14.808}{593 \times 14.7}} = 544{,}000 \text{ cu ft of free vapor per hr at 68 F}$$

(c) Estimated time required to reach the boiling point at the observed (20,000 Btu) heat-absorption rate is

$$\theta = \frac{D\rho C_p(T-70)}{4I} + \frac{4.87\,t(T+20)}{I} \ldots\ldots [11]'''$$

$$\theta = \frac{30 \times 49.5 \times 0.51(133-70)}{4 \times 20{,}000} + \frac{4.87 \times 0.375(133+20)}{20{,}000}$$

$$\theta = 0.5965 + 0.0140$$

$$\theta = 0.6105 \text{ hr} = 36.6 \text{ min}$$

2 Assume that the same vessel has been emptied and is then surrounded with intense fire:

(a) The shell metal would reach 1100 F in about

$$\theta = 0.320\, t = 0.320 \times 0.375 \ldots\ldots [31]$$

$$\theta = 0.12 \text{ hr} = 7.2 \text{ min}$$

(b) After the heating period, the required size of vent to discharge the expanding vapor at 3 in. water pressure would be

$$d = \sqrt{\frac{0.0407\,S}{\sqrt{P}}} \ldots\ldots [6]'''$$

in which

$$S = 3013 \text{ sq ft}$$

$$d = \sqrt{\frac{0.0407 \times 3013}{\sqrt{0.1083}}} = 19.3 \text{ in.}$$

The necessary capacity of the relief device would be

$$Q = 1550\, d^2 \ldots\ldots [26]^{\text{iv}}$$

Q = 578,000 cu ft of free air per hr at 68 F

Q = 9620 cu ft of free air per min at 68 F

Discussion

T. A. Gadwa.[35] The proposed constant heat-input rate of 20,000 Btu per sq ft per hr is considerably higher than the "Wartime Recommendations of Safety Valve Standardization Conference," which also considers a variable rate, depending upon the total surface of the vessel. The concept of wetted surface is a logical basis for calculation of exposed surface. Since the heat rate and wetted surface are the major factors affecting the relief-valve size, it would be desirable to have the conference review their recommendations with respect to these latest data.

The suggestion in the paper of considering the relief-valve connection as a short tube orifice is questionable. In order to obtain a 0.97 coefficient, a tapered nozzle is necessary, so that the inlet connection is always larger than the orifice, and both are established by the manufacturer. It is important always to connect the relief valve directly to a nozzle on the vessel rather than by means of a length of pipe, otherwise the pressure drop in the line will limit the capacity of the safety valve.

It is interesting to note that the code equation does not hold for pressures below approximately 15 psig and a different equation is necessary. When a relief valve operates at sufficiently high pressure, application of the gas-law deviation factor to the gas density will reduce the required orifice area. The code equation does not include this factor, although its use is suggested by the conference. Here again, possibly as a wartime measure, the API-ASME code could be revised to include the gas-deviation factor.

The heat rate for a vessel with insulation would be very useful since most fractionating towers would be in this category. Inasmuch as the recommended procedure requires orifice sizes larger than present methods, the question of whether test conditions duplicate the actual conditions must be considered before revisions are in order.

A. B. Guise.[36] The writer was especially interested in Part 1 of this paper, Observed Rate of Heat Absorption, because this is not only the key to the entire problem of adequate relief vents, but the subject of greatest controversy. When a member of the Flammable Liquid Committee of the NFPA, the writer became interested in the derivation of the tables of emergency-relief vents for tanks of various sizes and examined into the origins of these tables. As the authors mentioned, it was found that the only heat-absorption rate given was that of 6000 Btu per sq ft per hr, which applied only to large vertical tanks. In the course

[35] The Lummus Company, New York, N. Y.

[36] Vice-President, in Charge of Development, DuGas Engineering Corporation, Marinette, Wis.

of this committee's work, information was obtained from the Standard Oil Company of New Jersey on exposure fires around 1000-gal tanks in which heat-absorption rates of 24,000 Btu per sq ft per hr were obtained. These tests were not mentioned in the paper, so the assumption is that the authors were unaware of them. However, the figures check excellently with those found in the tests described in the paper.

Subsequently, the writer became curious as to the accuracy of the emergency-relief-vent tables when applied to pure compounds, such as benzol, alcohol, and acetone, and derived an equation similar to Equation [9] of the paper. However, simplification of this equation was never undertaken by the use a "contents factor," with which the authors have done such an excellent job of making this equation more readily usable.

It is noted that the vessels are assumed to be 90 per cent full of material when exposed to fire. In the writer's calculations, the same assumption was made, and he agrees thoroughly that this is the best condition to use for computations.

All of us who have struggled under the necessity of using our "best judgment," in place of the authoritative information and thoroughgoing analysis of the subject as given in this paper, will be pleased to have a logical basis upon which the fire-protection engineer can attack his problems.

About 2 years ago such information would have been invaluable to the writer, when he was informed that a certain organization recommended the use of 3000 Btu per sq ft per hr as the heat input to a 38-ft-diam butadiene sphere. In his calculations, the value 12,000 Btu per sq ft per hr was used and fortunately his advice was followed in this case.

F. L. Maker.[37] The authors have evidently done a very great amount of work in preparing this paper. Usually the results of so much labor should entitle the authors to some amount of grateful commendation from those whose future labors would be eased by the presentation of fundamentals completely worked out. In this case, however, it would appear that they brought most of the work on themselves by the fact that an essentially simple problem has been unbelievably complicated. The resulting recommendations, which do not necessarily follow from the facts presented, are in a form that is unsatisfactory from several points of view.

Time has not permitted preparation of a complete discussion of the entire paper, and the following remarks are concerned with the first two parts of the paper only, i.e., relief for vessels under pressure over 15 psi.

The test data presented in the first part include a description of tests run by Fetterly, the Underwriters Laboratory, by the Aluminum Company of America, and by the authors, in regard to maximum rates of heat absorption. As conclusions from these results, the authors recommend that the heat-absorption rate of 20,000 Btu per sq ft of wetted surface be adopted as the criterion for determining the relief capacity.

The authors recognize that this heat-absorption rate appears high when compared to over-all absorption rates on boilers and tube stills and point out that much higher rates are experienced in combustion chambers of radiant absorption sections of boilers. They are correct in regard to the much higher radiant-absorption rates in boilers; and even in oil stills rates in excess of 20,000 Btu per sq ft per hr on the surface most directly exposed to the fire are not uncommon. A point that was overlooked is that in oil stills the absorbing surface is deliberately arranged to limit the maximum heat-absorption rate to avoid coking or overheating of the oil on the inside of the tubes.

That such rates of heat absorption are possible is not new. The authors make reference to the Liquefied Petroleum Gas Safety Orders of the California Industrial Accident Commission. These were prepared about 9 or 10 years ago, and the writer suggested the particular form of the provision for safety valves that was adopted. A memorandum discussing the particular provision was presented at the time. This included the following:

"One of the most common emergency situations, particularly in the oil industry, is concerned with the possible exposure of a volatile fluid to fire. Tanks containing gasoline, casing head gasoline, butane, etc., may develop rather high vapor pressures at fairly moderate temperatures. In the case of a fire surrounding the vessels, which may occur due to breakage of lines, or perhaps from broken connections or leaks in the container itself, heat may be absorbed through the walls of the container that will cause vaporization of the volatile contents. In order to determine what amounts of vaporization might have to be taken care of, tests were run by the General Engineering Department of the Standard Oil Company of California some years ago (about 1925), in which a horizontal tank of water was surrounded by a fire in a pool of gasoline inside a dike and the quantity of vapor generated was measured. Heat-absorption rates were found as high as 25,000 Btu per sq ft of wetted surface; the tank being approximately half full of water."

The authors' 20,000 Btu per sq ft per hr would not be out of line for vessels completely exposed to fire. However, it does not appear reasonable to apply this same rate in all situations. The tests cited by the authors were in some cases run under conditions where the tank was pretty well enclosed. This condition might easily occur in a building, and for such cases the suggested rate would appear reasonable. The authors are apparently also interested primarily in vertical tanks. Most of the storage of products of high volatility in refinery and natural-gasoline processing plants, where the storage is within the plot limits of the plant, is usually in long horizontal tanks, running from 6 to 14 ft in diam, and lengths up to 100 ft. For such tanks the probability of the entire surface being exposed to maximum heat-absorption rates is small, particularly where they are installed in the open air and not inside of a building. Consequently, some deduction in average heat-absorption rate would appear to be reasonable. The suggestion made in connection with the California Liquefied Petroleum Gas Safety Orders was based on the following:

"For small containers, which are much more likely to be surrounded by fire, this maximum rate would probably be used as a basis for relief capacity. For larger containers it is unlikely that the entire surface will be surrounded by fire, so that lower rates appear reasonable. For butane tanks of sizes in the neighborhood of 1000 gallons capacity a rate of 20,000 Btu per sq ft per hour on 60 per cent of the total area of the tank (equivalent to 12,000 Btu per sq ft per hour on the entire surface of the tank) has been proposed. On very large tanks, and on those in which exposure on the entire tank might appear impossible, smaller rates yet may be used. In the case of butane tanks installed away from a source of fuel, except for a leak in piping, a rate of 25 per cent of the foregoing amounts was recommended. This amounts to 3000 Btu on the entire surface of the tank. D. V. Stroop, secretary of the API Fire Protection Committee, has proposed a somewhat similar condition in the form of a curve, which when plotted on logarithmic paper became almost a straight line. Making the substitution of a straight line leads to a formula of the following form

$$\frac{Q}{A} = \frac{48,000}{A^{1/3}}$$

where

Q = total amount of heat absorbed by vessel per hour
A = total area of vessel

[37] Designing Engineer for Refinery Plants, Standard Oil Company of California, San Francisco, Calif. Mem. A.S.M.E.

"The total heat absorbed by the vessel is therefore
$Q = 48{,}000\, A^{2/3}$ Btu per hr"

This formula is admittedly arbitrary. On the other hand, it would appear more reasonable for large storage vessels than using the same rate of heat absorption regardless of size. If the vessel is so enclosed that it is possible for it to be entirely surrounded by fire, which implies a limited size, the use of a maximum absorption rate of 20,000 Btu per sq ft does not appear unreasonable.

In the case of the California Liquefied Gas Safety Orders, the formula was further simplified, because only two commercial products, namely, commercial butane and propane fuel, were being considered, and the corresponding pressure conditions for vessels constructed under the safety order were limited.

From this point on, the paper becomes more difficult to follow. The objective of the authors is not quite clear, nor is it clear just what provision they are finally recommending. They suggest that their recommendations be embodied in official codes. Before this could be done, however, the presentation would have to be considerably simplified and clarified, and the various suggestions involved separated, so that they could be evaluated on their own merits.

After considerable study of the text and the derivation of formulas in the Appendix to Part 2, it appears that what the authors are proposing is to determine the size of a nozzle to put on a vessel. They also seem to be concerned with the fact that after a pressure vessel is built, it is frequently found later to contain other than the product it was originally designed for, and they apparently have in mind that the so-called, "contents factor F," will in some way give them a nozzle that would be large enough for a number of different products.

In addition they superpose a further suggestion that these nozzles be designed in sizes only for a limited number of pressures, namely, 15, 60, 150, 250, and 500 psi.

The entire problem involved would appear to be much simpler than the amount of mathematics presented in this paper seems to indicate. It seems to involve only the following steps:

1 Determine the amount of heat absorption to be provided for.

2 Determine the maximum pressure at which the discharge through the emergency relief device is to be computed, and the corresponding temperature.

3 Determine, for the particular product concerned, the number of pounds per hour that would be vaporized for this total heat absorption.

4 Choose a suitable emergency-relief device having a capacity sufficient to discharge the amount of vapor thus computed.

There is no advantage in attempting to accumulate all these steps into one all-embracing formula, and particularly to do this only for a limited number of pressure ranges. Rather, such an over-all formula tends to hide the results of the individual steps and renders any check of their reasonableness difficult.

In regard to the total amount of heat to be absorbed, the rate of 20,000 Btu per sq ft per hr on a wetted surface appears reasonable for small containers that can be entirely surrounded by flame, or even for fairly large vertical containers that may be so enclosed that they are likely to be entirely surrounded by flame in case of fire. For containers installed outdoors, some reduction is reasonable to take care of the unlikelihood of an entire vessel of large size being surrounded by an intense fire. The relation previously given, as used by the API, appears reasonable. It is admittedly arbitrary and might be improved upon. It is, however, convenient to use, it is qualitatively in the right direction, and in view of lack of evidence to the contrary, it is suggested that it continue to be used.

The next step is to determine the pressure at which the relief should be computed. This, of course, depends indirectly upon the product concerned, in that the pressure vessels presumably are made strong enough to hold the contents under normal conditions, with some leeway above the vapor pressure that would be reached in normal operation to prevent frequent poppings. Actually the vessel will be stronger than required to maintain the product at the vapor pressure corresponding to its normal temperature. The vessel will usually have some additional thickness, due to the use of commercial thicknesses of plate, and while relatively new, to the provision of some corrosion allowances. In the case of vessels for fluids of low vapor pressures, such as less than 50 psi at ordinary operating temperature, it will generally be found that the minimum thickness for which it is desirable to make the vessel will give a safe working pressure that may be considerably higher than the normal vapor pressure of the contents. In this case economy will be achieved by taking advantage of the maximum working pressure of the vessel and designing the emergency relief for this pressure. The corresponding temperature can be obtained from a vapor pressure-temperature curve of the product, which will either be available or can be approximated by general rules, such as a line drawn on a Cox chart from a given boiling point at atmospheric pressure. A safety valve should be set at this maximum allowable pressure, and if this is the only provision for emergency relief, its capacity should be computed on the basis of whatever "accumulation" or increase above the initial popping pressure is permissible for the maximum discharge rate. The A.S.M.E. Unfired Pressure Vessel Code allows 10 per cent accumulation. However, it should not be overlooked that the emergency condition being provided for is hoped to be very unusual, and that there appears no good reason why a maximum accumulation up to the pressure at which the vessel has been hydrostatically tested (generally at least 50 per cent above the design pressure) should not be permitted. Even above this pressure the safety factor of the vessel provides further leeway before the vessel might be expected to explode.

If a safety valve is provided of such size that it can take care of only normally expected conditions, further emergency fire relief can be provided by some form of bursting disk. Bursting disks have the disadvantage, however, that they cannot be designed to have bursting pressures very close to the normal operating pressures and have a reasonable life. A bursting disk that will burst at a pressure twice or 1.5 times the normal maximum pressure can be expected, however, to have a reasonable life. If the normal operating pressure is close to the design pressure of the vessel, it is suggested that bursting disks added for emergency fire exposure, in addition to a safety valve set at the design pressure, be arranged to have their bursting pressure at 150 per cent of the design pressure of the vessel.

For bursting disks, which are relatively cheap compared to safety valves, it is usually simple to provide plenty of capacity.

The purpose of the "contents factor F," used the the authors, is not at all clear. Partly, it seems to be introduced in order to reduce the determination of relief provision size to a single formula. The advantage of this is doubtful, as it succeeds admirably in hiding the reasonableness of each step in the process. Partly also, an impression is gathered that the authors proposed this factor because of some doubt that data on latent heat of vaporization would be readily available, and that by use of Fig. 12 it would be easier to guess the value of the "contents factor F" than it would be to guess the latent heat of vaporization. It is much more accurate to estimate the value of the latent heat than to choose the correct value of the more complicated value F.

Time has not permitted checking the curves of Fig. 12 to see what provision the authors have included for variation of the

latent heat of vaporization with temperature, nor is any table given showing how the various points were computed. This is one of the disadvantages of the use of an all-embracing formula, in that there is no ready way of checking the reasonableness of an important value used in the formula. The authors mention the use of Trouton's rule. Actually a curve based on Hildebrand's rule, that covers vaporization at other than atmospheric boiling points, should be used. Such a chart has been published.[38]

With the heat absorption, the discharge pressure, the temperature, and the latent heat of vaporization fixed, the quantity to be handled can be easily computed and the necessary size of the safety device chosen.

The dimensions of the relief device will depend upon what kind it is. The formula used in the API-ASME Code, due to Davidson and MacArdle, is devised primarily for hydrocarbons having a considerable number of atoms in the molecule. It will be on the safe side by about 10 per cent for a simpler molecule, such as ethane and propane, and other gases having few atoms in the molecule; these have higher values of the ratio of specific heats. The formula includes a coefficient of 0.97 for a tapered nozzle, because the originators of the formula worked for a company making a safety valve of the nozzle type, for which this coefficient applied. This same coefficient might not apply for other types of safety valves, particularly the low-lift type, previously in common use on boilers and air tanks.

The authors modify this formula to apply to a short tube having a coefficient of 0.83 instead of 0.97. At this point again, the reasoning becomes vague. It is not quite clear from here on what the authors mean to do with the diameter thus found. They talk about using the diameter thus determined for the "relief connection size," implying that they mean the actual nozzle attached to the vessel and not the nozzle that may be part of the safety valve, nor the diameter of any bursting-disk device. If this is what they have in mind, the nozzle so determined will be too small. The actual connection to the vessel in the safety valve or other relief device should be considerably larger in diameter than the limiting discharge orifice. If this is not the case, the entrance to the nozzle constitutes in effect a second orifice in series with the principal relief orifice, and when the flow starts, the pressure drop will not be almost entirely concentrated at the relief orifice, but will be divided between the relief orifice and entrance losses and friction in the connecting pipe between the vessel and the safety valve. This may cause the pressure at the safety valve to drop below the pressure at which it reseats, causing the valve to chatter. When a safety valve is used, the normal procedure is to choose the size of the safety valve from tables, furnished by the manufacturers, or to take the size a manufacturer recommends for the conditions, and then make the pipe connection correspond to the inlet flange on the safety valve. This will be usually several times the diameter of the actual nozzle in the safety valve. The discharge connection will be even larger.

The authors apparently also have the idea of making such computations only for a limited number of pressures. Their Fig. 12, for example, indicates pressures of 15, 60, 150, 250, and 500 psig, and they give four formulas, Equations [1]', [2]', [3]', and [4]', for particular pressures. Presumably they are trying to make it easy for an inspector or a designer to pick the nozzle sizes. Actually deciding upon safety-valve sizes by steps in this manner will cause considerable trouble. For example, Equation [2]' is suggested for working pressures from 60 to 200 psig. Presumably, therefore, it is based on 60 psig. If it is used to determine the area of a safety valve operating at 100 psig, it will require a larger-size safety valve than is actually needed. This may increase the expense very considerably in the case of the original installation, and even more if the rule is applied to an existing installation. A safety valve might be entirely adequate for the actual discharge pressure and yet not be large enough for the assumed lower pressure arbitrarily used for convenience. Such a provision is quite objectionable. The amount of time saved over computing the requirements for the actual operating conditions is inconsiderable, and the amount of extra expense in the installation may be of large magnitude.

As a general conclusion, the authors' suggestion that their recommendations be used as a basis for official codes probably will be emphatically disagreed with by most designers who have been using the API-ASME Code, because of its failure to allow for the less likelihood of exposure to fire of an entire area of a large tank, because of the lack of clarity of just what the authors are actually computing in their relief-device size, and because of the complicated way in which the result is arrived at, with the steps so combined that the implied values are not immediately evident for check as to their reasonableness.

[38] Ref. (12) of authors' paper, p. 13.

T. McLean Jasper.[39] While the writer does not feel competent to discuss this paper so as to add constructive information or criticism, it is easily recognized that this type of information has positive value in analyzing a very important problem. It supplies data which will tend to reduce the danger. The particular tribute the writer wishes to pay to the authors is the resort to a practical approach to the problem, and this after all is the superior method for any intricate matter.

The next step is to devise a simple workable plan for placing safety valves or disks and connections which will evacuate inflammable liquids from fires when they occur, thus rendering them less hazardous and thereby safeguarding working personnel and property.

F. L. Newcomb.[40] On January 4 and 5, 1943, a meeting of safety-valve manufacturers and users in the petroleum industry was called by the Petroleum Administration for War to draw up recommendations for wartime standardized petroleum safety valves, including their application. At that time, a heat-input curve was submitted to the Petroleum Administration and adopted for the duration. This heat-input curve was prepared by the Standard Oil Development Company and was a modification of the curve proposed by D. V. Stroop of the American Petroleum Institute. Both this curve and the one proposed by Mr. Stroop have been referred to previously. The Standard Oil Development Company curve was the result of recommendations from process engineers concerned with the design of furnaces and other heat-transfer equipment and was the result of long and careful consideration.

Through membership on the API Subcommittee on Relief Valve Capacities, there was available to the writer some data on actual fires which had occurred in the petroleum industry where the safety valves had prevented overpressure during a fire. By assuming that the safety valves blew to capacity and knowing the characteristics of the contents of the container, it was possible to figure the maximum heat input per square foot of exposed area. If the safety valves were not discharging at full capacity, the heat input would have been less than computed, and therefore the assumption was on the safe side. When these calculations were made and plotted against the curve submitted to the Petroleum Administration for War, in every case the maximum heat input was less than required by the curve. The matter of heat input and numerous other features are under consideration

[39] Engineer, A. O. Smith Corporation, Milwaukee, Wis. Mem. A.S.M.E.

[40] Senior Specialist, Standard Oil Development Company, Elizabeth, N. J. Mem. A.S.M.E.

by the API Subcommittee, and this curve has also been submitted to them as a proposal for a starting point in their work. It is possible that when time permits this committee to become more active and more data are obtainable, it may be advisable to modify the curve. It has, however, been in use for a number of years and there has never been any evidence that it has not been adequate. The lack of failures due to fire exposure of vessels reasonably protected with safety valves in the petroleum industry indicates that the present practices are adequate. It is felt that the recommendations contained in the paper under discussion would require unreasonably high safety-valve relieving capacity and place an entirely unnecessary burden on pressure-vessel users.

It is not felt, however, that the work done by the authors preparing this paper should, in any way, be overlooked in further studying the matter of safety-valve relieving capacity, but should be given due consideration along with all other data and experience which may be available.

R. C. Werner[41] and S. T. Russell.[41] This discussion has been limited to Part 1 on "Observed Rate of Heat Absorption." Each of the four groups of experiments was studied in turn and certain points were noted which were thought to have a vital bearing on the acceptance of a 20,000 Btu per hr per sq ft heat-absorption rate as a basis for sizing relief valves for vessels exposed to fire.

In regard to the test reported by J. F. Fetterly, Bureau of Explosives, 1929, the authors' use of 5.9 min as the time factor in calculating the rate of heat absorption of 23,300 Btu per hr per sq ft of wetted surface may be open to question.

It is reported by Fetterly: "It was intended that the safety valve should open to the full extent of its relief area, and when once opened to that area it should remain open until the contents of the container were discharged without exceeding 375 pounds per square inch gage pressure."

The irregularities in the pressure curve in Fetterly's publication may be explained entirely by Venturi effect, if the piping was arranged as in Fig. 3 of the paper under discussion, and the piping was of certain sizes. Fetterly's paper does not give details of the arrangement of the equipment. If it were assumed that the relief valve, once open, did not shut, the heat-absorption rate would have been 11,850 instead of 23,300 Btu.

It is plausible to reason that the temperature curve reported by Fetterly represents the condition at the bottom of the cylinder, and that vaporization might be taking place at the surface in spite of the temperature at the bottom being less than the theoretical boiling point of propane. If the propane contained some lower boiling material this effect would be more marked.

If one assumes that the mechanism outlined by the authors is correct the rate of heat absorption would vary as follows:

Time, min	Rate, Btu per sq ft per hr
0 to 3.7	1530
3.7 to 9.4	8410
9.4 to 15.3	23300

Fetterly's report does not indicate any physical basis for such a wide variation in rate of heat absorption.

In general, it may be said that it appears difficult to arrive at a correct value of the rate of heat absorption on the basis of the test data in Fetterly's publication.

In regard to the test made by the Underwriters' Laboratories, Inc., as reported in N.B.F.U. Bulletin of Research No. 3, 1938, the following three points are noted:

1 The introduction of a water film on the surface of a plate on the fire side increases the heat absorbed from the fire. A second test was performed with identical conditions except that no water film was present. From the dimensions of the vertical plate and the rate of the average temperature rise, the maximum rate at which heat was absorbed by the plate without the water film was calculated to be 461,000 Btu per hr, as compared to 776,000 Btu per hr with the water film. These results bring out the fact that if the quantity of heat absorbed is measured by a water film on the outside of the tank a considerably higher result will be obtained than for the case where no water film exists. This should be taken into account when applying other test data.

2 A considerable error may be introduced if one applies the results obtained from a small-scale test to commercial installations. As the oxygen supplied to burning fuel is increased to the theoretical quantity, the flame temperature increases rapidly. Therefore a fire with excess oxygen will generally be hotter than where a deficiency of oxygen exists. A small fire may well be the hotter because of its size, since an excess of oxygen over that needed for complete combustion can be easily obtained, but in the case of a fire of large area, it is difficult for the center portion to obtain any oxygen at all.

3 The size of the tank should also be considered. Although the actual initial flame temperature in two different cases may be identical, the larger tank will absorb a greater portion of the total heat in the flame and thus have a lower average flame temperature adjacent to the tank than for the other case. Small tanks and areas will, therefore, receive heat at a greater rate than large tanks or areas.

In regard to the report of the 1930 tests of the A.P.I. on aluminum-alloy tanks for tank trucks, the following points are noted:

1 To assume that a measure of the total heat supplied to the gasoline is the amount of heat needed to vaporize completely the gasoline may result in considerable error, particularly in the case of small tanks. Gasoline may leave the tank in the form of a liquid as entrainment. This can be gathered from the original data on this test. The vapors from the tank ignited at 4 min 20 sec after the start of the fire. At 6 min the fusible plug in the fill cap melted and atomized gasoline was observed to issue from the opening for several seconds. Since both openings are in the same cap it is possible that gasoline was atomized (leaving as a liquid) for the full period of 1 min 47 sec through the vent and probably longer but at a reduced rate.

2 To assume that the liquid level in the tank can be measured by the abrupt temperature rise as recorded by the thermocouples located at different elevations may result in error. The thermocouples were located on lugs on the outside surface. The temperature of the liquid on the inside of the tank would differ by the temperature gradient needed to transfer the heat from the outside surface through the metal walls and inside film to the liquid. This temperature gradient would vary from 30 to 200 F. With such a temperature gradient the inside temperature may be related to the thermocouple readings, but it would be difficult to determine the exact relationship.

On checking the original data, a difference was discovered between the recorded temperatures in the table and a plot of the temperatures. The value for the thermocouple No. 10 at 14 min is recorded as 355 F but plotted as 255 F. Replotting the temperature curve for thermocouple No. 10 with the correction given, it would appear that the abrupt temperature rise for both the No. 10 thermocouple (at 50 per cent level in the tank) and No. 11 thermocouple (at 75 per cent level in the tank) occurred at approximately the same time, i.e., 12 min from the start of the fire. Since both thermocouples rose abruptly at the same time, they do not seem to have given a very good indication of the liquid level.

[41] Blaw-Knox Construction Company, Pittsburgh, Pa.

A reasonable explanation may be obtained for this abrupt rise in temperature from the photographic record of the test in the original report. The photographs as a rule were taken every 3 min. At 9 min there was only a slight indication of any wind blowing. This could be observed from the burning vent gases. At 12 min a slight breeze was blowing the flame from the burning vent gases toward the end of the tank on which the thermocouples were mounted, which were those reported in this paper. The thermocouples on the second tank on the opposite side at no time indicated such a sharp rise although the tank was only one half full to start and was exposed to the same fire. It will be noted from the photographs that the flame was blown away from the second tank by the wind. The abrupt temperature rise, therefore, seems to be dependent upon whether or not the thermocouples were exposed to the radiation from the burning vent gases, and not primarily to the liquid level in the tank.

3 To assume, after rapid ebullition is started, that the only heat needed is for evaporation seems to be an error. From the properties of the gasoline used in the test the initial distillation point is 95 F, and the 51 per cent distillation point is 284 F. The heat needed to evaporate the gasoline is that heat required for the latent heat of evaporation plus that needed as sensible heat to raise the temperature of the remaining gasoline to a rising boiling point.

4 The data on the other 150 gal tank should have been considered. This second tank was directly attached to the first and was exposed to the same fire. The charge consisted of only 75 gal of which 30 gal remained after the test. In all, only 45 gal were vented as compared to 129 gal for the tank reported in the paper under discussion. This marked difference cannot be explained by the difference in exposed area only. Additional explanation may result in the consideration of entrainment. The half-full tank would have on an average a lower liquid level thus resulting in less entrainment.

5 In applying the results of these tests to the design of a relief valve to be attached to a storage tank located in the open, a correction factor should be used with the test data since a windshield was used. Theoretically the radiation from the shield to the tank may be almost as great as from the flame alone. Windshields also help the natural convection of the fire and increase the per cent air to a possible excess. In the case of burning gasoline spread over a large area, there would be a deficiency of air, and therefore a much lower flame temperature.

In regard to the test made by the authors of the paper under discussion, the following points are noted:

1 It is questionable whether the asbestos shield around the tank approaches the condition of a flame of considerable depth. The emissivity of asbestos is higher than can be reached in a flame, and a flame of considerable depth probably would be of a much lower temperature, due to the difficulty of oxygen reaching all parts of the flame. Therefore the heat-transfer rate with the asbestos shield would be higher than could be obtained by a tank surrounded by a large-area fire. The authors were aware of this fact, but believed that the strong updraft between the cylinder and the tank tended to offset the increased radiation.

2 However, it seems questionable whether the strong updraft between the cylinder shield and the tank has the indicated effect. This strong updraft increases the natural convection to the tank and probably causes an increase in the air supply giving a more theoretical mixture and thus a higher flame temperature, as compared to a large-area fire which would have a deficiency of oxygen in many places.

3 On checking the information presented for runs Nos. 2 and 3, it appears that the heat which the authors designate as latent heat absorbed is not a measured quantity but rather the difference between the actual measured sensible heat absorbed and the calculated theoretical possible heat absorbed. Therefore, the calculated rate of heat absorption is actually a theoretical value and should not be confused with the values reported in runs Nos. 1 and 4, which are actual measured rates.

4 The results obtained in run No. 4 are questionable due to the reduction in area used in the calculation by the assumption that a band of the wetted surface 2 ft wide is not exposed to the flame. By using the full wetted area a value of 12,630 Btu per sq ft per hr is calculated. If the authors' assumption is used, a value of 16,850 Btu per sq ft per hr is obtained which is 33.4 per cent higher. Since the actual measured flame temperature is higher than for runs Nos. 2 and 3, it is questionable whether such an assumption which results in a 33.4 per cent increase should be made without at least more test data.

5 As was pointed out in regard to the test made by the Underwriters' Laboratories, Inc., and which is observed when comparing runs Nos. 1 and 4 (25,900 as against 16,850 Btu per sq ft per hr) the tank or plate with water flowing on the surface facing the flame will absorb heat at a considerably higher rate than if the water film is omitted. This would indicate, as mentioned before, that a water film increased the absorption rate and therefore such test data cannot be applied to a dry tank.

6 In calculating the results of run No. 1, the authors have assumed a mechanism whereby the tank metal receives the radiation and then transfers the heat to the water film. This assumption is open to question. The measure of absorptivity of a material to radiation is its emissivity. McAdams reports the maximum emissivity of iron and steel at around 0.75, whereas for water a value of 0.96 is reported. This would indicate that the water film is more absorptive to heat radiation than steel.

Underwriters' Laboratories, Inc., reports in its test; "A simple experiment to gage the properties of water film may be performed by viewing a flame through a thin sheet of glass. The difference in heat energy sensed by the face or hand with or without the glass is not appreciable; but when an unbroken film of water is caused to pass over the glass surface the difference in the heat transmitted through the combination is quite perceptible, although translucency is not materially altered." However, in the authors' test the tank metal became hotter than the water film. This appears to be in contradiction to Underwriters' Laboratories, Inc., observation and to theory. More data along this line are therefore needed.

In summarizing there seem to be three experimental procedures of major importance which should first be evaluated before the results obtained by using such procedures are adapted to a relatively large storage tank located in the open and heated by flame such as a large-area oil fire.

1 The use of a water film on the outside of a tank to measure the quantity of heat absorbed. The water film in two of the reported cases greatly increased the amount of heat absorbed.

2 The use of the quantity of material which is vented from the tank as a measure of the quantity of heat absorbed when it is assumed that the material was vented as a vapor. In one particular case, atomized gasoline was observed to issue from the tank.

3 The use of a windshield to control the fire. The emissivity of the shield is generally higher than the possible flame emissivity and thus greatly increases the radiation to the tank. In all the reported cases some type of windshield was used.

In conclusion, the authors of this paper should be congratulated for their contribution to the field of design and safety. It is good to see that even during a period of accelerated production the field of safety has not altogether been forgotten.

Walter Samans.[42] Because of lack of time for adequate study of this paper, it would be an unwarranted presumption on the writer's part to make any off-hand comments on the authors' findings. However, their correlation of formerly existing test data with theory indicates that more tests were very much needed to confirm the theories. The chemical industries particularly will appreciate this report, and the authors' employers, who fostered the tests, in absorbing the costs for the equipment and personnel engaged, are entitled to commendation jointly with the authors themselves. The data thereby presented should prove the best possible source for the formulation of a more standardized practice in determining the required safety valves supplemented by relief disks or cups, and similar devices.

The writer's comments will apply principally to large vertical storage tanks. The authors have recognized a difference between test fires around small pressure vessels and fires in commercial plants where the hindrance offered to complete envelopment by flame will materially cut down the total heat absorbed. They have acknowledged this difference in Part 1 of their paper for the test reported by Fetterly, and those conducted for the A.P.I. and N.F.P.A. They have acknowledged the justification for a reduced heat input on vertical storage tanks over 15 ft diam, approaching a rate of 6000 Btu per sq ft per hr for a 100,000-gal-capacity tank, and reaching the maximum 24-in. size of relief vent for tanks of 300,000 gal (7000 bbl approximately) and arger, in Part 3, Fig. 18.

Although this appears to be a concession to the experience in actual tank fires, and purports to be a reasonable interpretation of test results, this may not prove a sound reason for discarding the present practices for petroleum storage tanks, as shown by curve 6 of Fig. 18. This stand is taken both because of the character of the hydrocarbon flame encountered when pools of petroleum, as crude, reduced crude, intermediates, or gasoline are on fire in a tank pit, and because vertical storage tanks do rupture at the edge of the roof without destructive explosion.

Assume that burning oil on the ground is a heavy petroleum product and that there is no appreciable wind (rain is not a consideration); there will be lots of smoke which screens the tank from radiant heat, except where the inrushing air enters the vapor. Inside of that boundary there is incomplete combustion, because of insufficient air, and consequently a lower temperature. Above the top of a tank 30 ft or more in height, the radiant heat has little effect on the roof. It is quite possible that the equivalent of only 1/6 of the wetted shell surface need be computed at a heat input of 20,000 Btu per sq ft per hr. In the case of burning gasoline, the flame may be almost free of smoke, or at least will appear so on the outer boundary of the burning cloud of vapor. There could not be a sea of fire over an area having an annular width equal to the radius of a large tank that could be "seen" by the wetted tank surface as the interior of the burning vapor cloud is not clear. Smoke does form.

Because of the greater area on fire, the inrushing air will carry the flame higher, but by attenuation of the flame, the radiant-heat effect will be reduced with height, radiating less total heat per average square foot on higher tanks, and to the tank roof. Intermediate oils, between heavy fuel and gasoline, will show "in-between" effects.

When there is a wind blowing, there is no great difference in rate of heat input until its velocity is higher than the air drawn to the fire by the draft. There can never be the chimney effect produced by the authors' tests, because there the gasoline was sprayed into the rising current of air to produce a hot continuous flame with resulting excess air, as far as the photographic cuts indicate.

To simulate a storage-tank fire, tests might be made by burning gasoline poured on top of a pool of water around the tank The screen or shield should be placed outside of the basin that holds or receives the cooling water, and air admitted through a sufficiently large clearance under the screen. The effect should be similar to that obtained in the trial runs using gas fuel, shown in Fig. 8, except that, gasoline vapor having a higher molecular weight, and the flame being screened, a higher rate of heat input may be expected.

In this connection the authors' "unfortunate experiences" with small tanks should not affect conclusions based upon good experience with large vertical oil-storage tanks. To evaluate the screening effect of the width (i.e., horizontal depth) of flame on the heat radiated to the surface of the tank, the Underwriters' Laboratory test made on a flat plate might be reproduced, but shielding the boundaries of the plate from flame contact so that the air-vapor mixture will not take place near the test plate. Obviously, the results should give a low rate of heat input, when a heavy fuel oil is burned and not as high a rate as on the authors' test recorded in the paper, when burning gasoline.

The statements just given are also intended to verify the authors' conclusion that open-air combustion cannot be compared with controlled combustion in a boiler furnace. In the latter, only the wall tubes and radiant-heat surface are comparable, but the type of furnace flame, with little obstruction, and that due to haze, is seldom reproduced in industrial fires. It is expected that the subject of vertical storage-tank fires, covered by this contribution to the discussion, will be amplified by the actual experience of some of the many oil men in California who have combated fires in oil refineries and field storage tanks. The record so obtained should be combined with the results of the authors' studies and tests and formulated into the best possible procedure for determining the protective venting needed for large tanks.

Another point which appears to need clarification is the statement in the Summary at the beginning of the paper: "So far as can be learned, there is no substantial theory or evidence to support the limiting of heat-input rates to any particular height. There are some indications to the contrary, that is, high flame temperatures apparently exist at relatively great heights above the fuel during large fires."

There appears no reason for this reversal of published "Fire Fighter" opinion, i.e., not by the text of the authors' paper, and there being no other publication referred to in this thesis, possibly the authors will say why this statement was made in the Summary of Part 1, where reported tests did not develop "relatively great heights," unless other records made and cited in the original reports were not herein reproduced. The wording of the statement referred to may be unintentionally misleading. "High gas" temperatures will exist wherever they are found, and what was no doubt meant was that high temperatures are radiated to relatively great heights, provided the fire is intense and large enough. As mentioned in this discussion on fires around oil tanks inrushing air over an annular flaming oil area surrounding a tank, "attentuates" the flames and, for the case herein considered, will hasten the completion of vapor combustion. Obviously, the larger the area covered by fire, the higher the flame will reach, and the greater the intensity of radiation in the most favorable directions. Reverting to the original statement on "the limiting of heat-input rates to any particular height," the authors' tests did not simulate an open-air fire and on such fires the radiation effect of flames above a pool of oil on the ground will diminish rapidly upward as the intensity decreases.

[42] Sun Oil Company, Philadelphia, Pa.; member of the A.S.M.E. Boiler Code Committee; Chairman of the API-ASME Committee on Unfired Pressure Vessels; Vice-Chairman, API-Oil Storage Tank Committee. Mem. A.S.M.E.

The authors' theories are presented in a usable manner by the formulas, and the derivations are of practical form. They seem well substantiated by the tests. The authors have also recognized the varied degree of exposure encountered with special protection that may be provided in the construction. It now remains for interested authorities, as represented by state inspectors, insurance engineers, and recognized technical committees to formulate reasonable assumptions of departure from test conditions, to which the authors' correlation between tests and theories may be properly applied to give adequate protection for each type and size of vessel or tank.

The authors' work was well worth while. The resulting forward steps have anticipated similar work by other organizations that was no doubt postponed because of the existing war emergency. The authors' continued work for the furtherance of their ideas should be aided and encouraged in all ways possible.

H. R. Zeigler.[43] The derivation of the fundamental equation is very simple and logical. The authors start with the formula given in the API-ASME Code which is used for determination of capacities of pressure-relieving safety devices. They claim that this formula is universally employed and is of proved accuracy. By substitution, factors for heat input, wetted surface, and latent heat are introduced to complete the formula.

A novel feature of the paper is a chart which shows "contents factors" for many compounds. This chart may be read and the reading substituted in place of one expression in the formula which involves temperature of the substance, molecular weight, and latent heat of vaporization of the substance.

In addition to the basic equation, the authors offer a few other equations, the purpose of which is not particularly clear. However, all of these equations are extremely easy of solution, whereas the well-known Fetterly formula is quite difficult of solution and offers many possibilities of error.

One novel suggestion is that relief valves be made the same size on all vessels of the same capacity and pressure range. This is to promote interchangeability of use of such vessels and, consequently, a high "content factor" is used.

The authors concede indirectly that the size of the tank has an influence, if the writer interprets the paper correctly. This statement is based on the fact that they treat of four pressure ranges for vessels not exceeding 10,000-gal capacity, and one formula only for vessels above that size. However, another way of looking at this is that they may consider it uneconomical to bother with different relief areas for vessels of many differ nt working pressures. The four pressure ranges are 15–60, 60–200, 200–700 psig and finally, pressures above 700 psig. Incidentally, in O. M. Setrum's study of the subject of relief valves, he suggested that the external temperature of tanks of 10,000 gal and larger should be 1000 F, instead of the 1200 F used for smaller tanks.

In leading up to the equation recommended, the authors exert all their efforts to establish a heat-input rate of 20,000 Btu per hr per sq ft of wetted surface. In reviewing previous work they point out that the API Committee on Fire Prevention prepared a chart 12 or 14 years ago for gasoline tanks exposed to fire. This chart shows capacity and size of relief devices for heat-input rates of 24,000 to 6000 Btu per hr per sq ft, but there was no practical basis presented for this choice of a heat rate. The relief standards contained in N.B.F.U. pamphlet 30 are based on heat input of 6000 Btu per hr per sq ft of wetted surface. They regard the work of Fetterly as outstanding and, therefore, discuss it at some length.

In discussing Fetterly's formula, the authors show that judged by modern methods of calculation, his coefficient of heat input was remarkably accurate. They then show that, since his estimate of surrounding temperature was 1200 F, the heat-input rate was about 14,000 Btu per hr per sq ft of wetted surface. However, they advance the idea that 1400 F is a more probable surrounding temperature and, therefore, according to Fetterly's coefficient the heat-input rate would be 23,000 Btu per hr per sq ft of wetted surface. The writer believes this disregards the fact that Fetterly's coefficient must have been obtained empirically from a calculated input rate and, therefore, the premise is untenable. It is also felt there were some other distortions of Fetterly's test data in the authors' analysis. Finally, the authors proved to their satisfaction that vaporization was in excess of relief-valve capacity in spite of the fact that the fire test on which Fetterly based his paper was satisfactorily performed.

The authors summarize their tests as follows: "When a tank is surrounded by an intense fire having an effective temperature of about 1400 F, heat is absorbed from the flames at a rate of the order of 20,000 Btu per hr per sq ft of wetted surface exposed."

The tests do not make out a very good case for either heat input of 20,000 Btu per hr per sq ft of wetted surface, or for the effective temperature of 1400 F. The tabulated results for three runs out of four were 25,900, 17,300, and 18,700 Btu and 1510, 1270, and 1315 F. These effective temperatures were measured at the center of the flame region under well-shielded conditions, and it is the writer's opinion that, unless a tank were installed within a building, windage would discount the effective temperature and, consequently, the heat-input rate very considerably.

The formula recommended by the authors seems to have only one weakness, namely, the heat-input rate which they have tried so hard to establish at 20,000 Btu per hr per sq ft of wetted surface.

The writer has worked out the following relief-valve areas for an 82.5-in-ID (7-ft-OD) × 42-ft T.L. tank. This tank was chosen because it has a nominal capacity of 10,000 gal and a total surface of 1000 sq ft. The working pressure is 200 psig, and this is chosen because it is a pressure very commonly used by the writer's company. This size and working pressure give an opportunity to show several different solutions for relief area.

Equation [2]′ covers vessels between 60 and 200 psig and is developed from Equation [4]′, being simplified by the insertion of 66 psia, and the wetted surface is expressed as 85 per cent of the total surface.

Equation [3]′ covers vessels between 200 and 700 psig and is developed in the same manner except that 220 psia is used.

Although Equation [4]′ will not be used, it is necessary to point out that this is based on the assumption that tanks should be equipped with relieving devices of sufficient capacity to promote interchangeability.

Equation [5]′ covers vessels larger than 10,000 gal. By inference the preceding equations are for vesseis less than 10,000-gal capacity.

Equation [2]′ $a = 0.0192S = 19.2$ sq in.
Equation [3]′ $a = 0.01112S = 11.12$ sq in.

$$\text{Equation [5]}'\ d = \sqrt{\frac{97.3S_w}{(P + 14.7)}\left(\frac{T}{Mr^2}\right)^{1/4}} = 2.715\text{ in.}\ldots$$

.... $a = 6.15$ sq in.
Fetterly's formula using 1400 F.......... $a = 4.91$ sq in.
Fetterly's formula using 1200 F.......... $a = 3.52$ sq in.
$D \times U$ table where $D = 7$ and $U = 45$; $D \times U = 315$ and $a = 3.10$ sq in.

In Equation [5]′ S_w = wetted surface, sq ft
T = temperature F absolute of vapor
P = gage pressure (R.V. setting pressure)

[43] Phillips Petroleum Company, Bartlesville, Okla.

M = molecular weight of vapor
r = latent heat, Btu per lb

The writer feels that the authors have not proved their constants sufficiently for adoption, in view of the success of the present practices recommended by the National Board of Fire Underwriters in Pamphlet 58.

The authors have not proved the inadequacy of Fetterly's formula.

In general, the authors offer a very intelligent basic formula for the solution of relief-valve-area problems and, if their constants for heat input and surrounding temperatures can be adjusted downward to produce more reasonable results, the writer would like very much to see this basic formula adopted.

T. C. Smith.[44] The authors of this paper are to be congratulated for this excellent paper with its summary of fire tests conducted by themselves and others. Particular attention should be directed to the method used for cooling the test tank and recognition given to this type of protecting against overpressure or overheating of the tank surface in the vapor space.

It has been difficult to understand why the authors have chosen to determine the pipe size required for the relief-valve connection rather than the orifice area of the valve. Commercial spring-loaded relief valves have an orifice area ranging from 25 to 30 per cent of the connecting-pipe area, hence it would appear that the orifice should be of primary concern and the inlet-pipe connection of secondary interest.

Also, consideration should be given to replotting the curves of "content factors" with the term "pressure" incorporated in the equation. This will permit a series of charts, each consisting of a family of curves for various groups of contents.

The authors present heat-input rates ranging from 20,000 to 30,000 Btu per sq ft per hr, as obtained from three different sets of tests. These values are not questioned. Next, they compare these results with established venting rules and create doubt as to their safety. Lastly, they call attention to the absence of specified heat-input rates in two well-known pressure-vessel codes, thereby inferring that this matter has been given scant attention by bodies interested in the formulation of safety standards.

The writer hastens to correct any doubts, on the deficiency of these venting rules, or impressions that this subject has been or is being neglected by groups engaged in the initial drafting of safety standards. These existing rules, although differing in form, employ definite heat-input values which have been established by experience. Also, these sponsoring groups receive many reports of tank and vessel failure which are studied to determine whether revisions in their standards are necessary. The heat-input rates and some of the current work being done is reported herewith as a matter of record.

The first venting rules, for the protection of equipment from overpressure when exposed to external fire, appear to be for oil-storage tanks operating at atmospheric pressure. The American Petroleum Institute was asked to suggest some kind of a venting schedule, and after reviewing the records of their industry where tanks were involved in fires, arbitrarily selected a unit-heat-input rule which was represented by their experience. This rule expressed mathematically is

$$Q/A = \frac{48{,}000}{A^{1/3}}$$

where

Q = total heat absorbed by tank per hour
A = wetted surface exposed to heat

[44] General Petroleum Corporation, Los Angeles, Calif.

The development of liquefied petroleum gases for commercial use brought out two venting rules intended for small- and moderate-size pressure tanks. The California Safety Orders for Liquefied Petroleum Gases gave a rule which was based on 20,000 Btu input on 60 per cent of the surface, or an average of 12,000 Btu per sq ft on the gross surface. The National Board of Fire Underwriter's rule was based upon a modification of Fetterly's formula, using a flame temperature of 1200 F for tanks having a capacity of 10,000 gal or less, and 1000 F for tanks of 30,000 gal capacity or more, with adjusted temperatures for tanks of intermediate capacity. The heat input by this rule will vary between 23,000 and 13,000 Btu per sq ft per hr.

The application of either the California or NBFU rule to extremely large vessels resulted in relieving areas that appeared to be excessive and caused them to be questioned. The American Petroleum Institute, because of its interest in pressure-vessel safety and fire prevention, became interested in the problem and referred the matter to a subcommittee for further study. This subcommittee was instructed to confine its initial efforts to large pressure storage tanks and determine if the previous A.P.I. rule for heat input to atmospheric tanks was satisfactory for pressure tanks, also what modifications might be necessary for other factors involved.

The assignment has been divided into three parts for study, namely, (*a*) the correlation of test data to actual fire conditions; (*b*) further tests, if necessary, to determine the scale effect for both horizontal and vertical vessels; and (*c*) a survey of fires involving pressure vessels to determine the maximum average heat input.

Some progress has been made on the last item. A questionnaire was issued to the petroleum industry for histories of external fires on pressure vessels. Cases have been selected where the vessel was full or nearly full of liquid hydrocarbons and both incoming and outgoing lines were closed. Such cases with pertinent detail data permit the estimation of the average heat input by assuming that the open relief valve is an orifice. This survey has produced heat-input data for two extreme cases as follows:

(*c*-1) More than 11,000, but less then 17,000 Btu per sq ft per hr, for small propane cylinders having a wetted surface of 16 sq ft. Approximately twenty-five cylinders exposed in ten or twelve fires failed from insufficient relief-valve area (0.062 sq in.). This experience prompted a program for changing the type of relief valve to one having a larger orifice (0.100 in.), and the results to date show that some twelve cylinders have successfully withstood severe fires. In at least two cases there were, in the same fire, cylinders with each type of valve, and some of those with the small orifice failed from excessive pressure while all of those with the large orifice came through without damage.

(*c*-2) Approximately 2300 Btu per sq ft per hr on a sphere having a wetted surface of 4363 sq ft. This was a 48-ft-diam sphere subjected to a severe fire for 1 hr 7 min. At times the vessel was completely enveloped in flame. The relief valves were adequate and prevented failure of the sphere.

(*c*-3) In addition to the foregoing, three intermediate cases can be cited as follows:

3600 Btu on a natural-gasoline tank; wetted surface = 319 sq ft
2050 Btu on a natural-gasoline tank; wetted surface = 900 sq ft
1350 Btu on a natural-gasoline tank; wetted surface = 990 sq ft

In all of these fires, the average heat input is estimated to be less than the value obtained by the previously mentioned API heat-input rule for oil-storage tanks.

Reports have been received on many other vessels, but unfortunately, the data do not permit an estimation of heat input because of open lines discharging liquid or vapor, or flame impingement on a surface in the vapor space which resulted in

overheating and failure of the metal. Such failures cannot be prevented by relieving devices which maintain, regardless of the size of the device, a fixed pressure on the tank.

Obviously, it would be both improper and premature to predict the final recommendations of this subcommittee.

As noted previously, there are existing rules for venting in use in the petroleum industry, and so far as is known, there is no evidence that these rules where properly applied have been inadequate to maintain safe conditions. The authors of the paper under discussion propose that these rules be superseded on the basis of the evidence and conclusions which they have submitted. In the writer's opinion, this is not justified by their data.

The experimental accuracy of their data is not being questioned, but the interpretation and extrapolation thereof on which their recommendations are based is open to criticism. They have employed results from a very limited number of tests run on a narrow range of vessel sizes under conditions which do not duplicate, nor in some cases resemble, actual service conditions. They have used these as a basis for a proposal that a satisfactory experience record be discounted and supplanted. This is unsound.

The data that they have derived justify conclusions only within the range of exposure conditions covered by the reported tests. However, if correlated with other accumulated data they can aid in reaching a reasonable solution for the over-all problem and for this purpose they are of full value. It is recommended that they be considered at this time only as a contribution to the study being made by the API subcommittee mentioned previously.

Authors' Closure

Reply to Discussion by T. A. Gadwa

Herewith are the authors' comments on the questions raised by Dr. Gadwa in his discussion of their paper.

Effective Height of Flame. Dr. Gadwa's reference to 50 ft for a minimum height for protection against fire exposures is the highest about which we have learned and with which we are in agreement to this extent: We have experienced heat failures (1000 to 1200F) of unprotected structural steel at a 50-ft elevation in the open. However, under such circumstances, a good part of the fire fuel was about 20 ft above grade. We have had such failures in the open at 30-to-40-ft above-ground spillage fires, and this was a part of the basis for the suggestions near the end of Part 1 of the paper. We are convinced that the more generally used height of 20 ft is inadequate.

Input Rate of 20,000 Btu per Sq Ft per Hr. At this time, we have a record of ten tests by six separate concerns, and five analyses of actual fires which indicate that the constant rate of 20,000 Btu per sq ft per hr of surface wetted by the contents is a safe minimum for vessels enveloped with intense fire. We do, however, suggest a reduction of this rate for atmospheric tanks above 20,000 gal nominal capacity, as proportionately represented by curve 2, Fig. 14, or curve 3, Fig. 18, of the paper. The authors interpret the "Wartime Recommendations . . ." to specify heat rates of 10,700 Btu for 10 sq ft of surface downward to 4800 Btu for 1000 sq ft per hr of wetted surface. In the light of our fire experience, this is inadequate, and no attempt will be made to co-ordinate our results with these recommendations.

Effect of Insulation. We suggest allowance for insulation in the capacity of the relieving apparatus but not in the area of the relieving connection. This would depend upon the reduction effected in the flow of heat by the insulation when exposed to hot gas temperatures of 1400F. No data have been prepared on this as it is expected that only in rare cases will allowances prove economical and practical.

Conversion of API-ASME Code Equation. To clarify the reason for the conversion of the API-ASME orifice equation, reference is suggested to its derivation. This will be found in "Discharge Capacity of Relief Valves for Oil Stills," by K. S. M. Davidson and D. W. MacArdle.[21]

It will be seen in this reference that the equation was derived using a ratio of specific heats of 1.001, a critical pressure ratio of 0.6064, and a coefficient of discharge of 0.97. These superficial or ideal values (the first two) were employed to assure adequate relief capacities for any of the heavier hydrocarbons. Since the formula is so universally employed, well known, and of proved accuracy, it was considered wise to incorporate it in the paper rather than those which we had derived for synthetic organics, even though there was little difference.

Further, it is our method to compute all relief capacities in terms of free area and then select the relief device of a capacity based upon approved flow test to pass the calculated rate. Since we use valves having capacities which range from 14 to 60 per cent of the capacity of the relief connection of the same size, it is frequently necessary to choose a larger relief connection than the calculated free area. We do not figure the relief capacity of the valve orifice itself because only those capacities for valves which are essentially free-blowing orifices (top guided, high lift, full bore) can be calculated readily with practical accuracy.

Also, it will be found expedient for most cases that the necessary relief device for fire exposure be supplemental to the usual spring-loaded safety valve, and that this device be simple, inexpensive, and of relatively large capacity. Such apparatus is available with free areas practically equivalent to the pipe-size connection. Reference is suggested to paragraph 5, Proposed Summary for Use, Part 2 of the paper.

Reply to Discussion by F. L. Maker

In order to simplify the authors' reply to Mr. Maker's lengthy discussion, a summary of his comments is given as follows:

1 Mr. Maker's comments refer to Parts 1 and 2 of the paper.

2 He states that heat-input rates of the order of 20,000 Btu per sq ft per hr are not new.

3 He suggests continued use of $\frac{Q}{A} = \frac{48{,}000}{A^{1/3}}$.

4 He wonders why we refer to nozzle size rather than relief-device size.

5 He is confused by the "contents factor."

6 He suggests how the problem should be solved.

All of Mr. Maker's comments in which relief devices are mentioned should be ignored because our paper is concerned with the free-relief area rather than the relief device itself.

The following reply is more to the points summarized:

1 Although Mr. Maker's comments refer to Parts 1 and 2, the authors wish to point out that a fire is no respecter of the working pressure of a vessel. Therefore, although the fire experience to which the authors refer consists of observed failure of low-pressure vessels up to 10,000-gal capacity exposed to fire out in the open, the intensity of a fire around similar high-pressure vessels would be the same. Reference is made by Mr. Maker to the difference between vessels in refineries and those in other plants. The codes do not recognize this differentiation and the authors are of the opinion that the code should be specific about the size and shape of a vessel in relation to heat input in an exposure fire, or else vessels should be designed for the worst condition. The formulas given apply to any shape and working pressure.

2 The authors are aware that possible heat-input rates of 20,000 Btu per sq ft per hr are not new but they desire to point out that very little information has been given to the public to this effect in the 18 years since heat-absorption rates of this

magnitude were observed by Mr. Maker. Furthermore, because of the use of recommended values of from one sixth to one third of the observed heat-input rates many a tank has probably suffered damage during this period.

3 Mr. Maker suggests continued use of the arbitrary expression $Q/A = \frac{48,000}{A^{1/3}}$ as a relation between heat input and surface area of a vessel. This is an example of arbitrary rule which the authors feel can and should be replaced by a more practical and accurate expression. The absurdity of this expression may be brought out by equating the right-hand side to 20,000 Btu per sq ft per hr and solving for A, the area of a vessel for which a heat-input rate of 20,000 Btu per sq ft per hr would be recommended; thus

$$\frac{48,000}{A^{1/3}} = 20,000$$
$$A^{1/3} = 2.4$$
$$A = (2.4)^3 = 13.8 \text{ sq ft}$$

This is equivalent to a vessel about 18 in. in diameter and 36 in. long. Our tests and fire experience indicate that 20,000 Btu per sq ft per hr may be expected up to areas of approximately 200 sq ft and 600 sq ft respectively. We, therefore, emphatically deny that this expression is reasonable.

4 The reasons for basing relieving capacity on the free-discharge area of relief connections are detailed in the reply to the comments by Turner C. Smith.

5 Mr. Maker is confused by the "contents factor." The authors feel that they have been quite clear in stating that this factor may be calculated for each individual liquid to be stored or processed and have suggested two excellent methods of estimating latent heats of vaporization. However, we also feel that by use of the recommended general expression for the "contents factor" in terms of pressure, one need not be concerned with the nature of material to be stored; and that not only is a factor of safety provided for, but also the cost of resulting vent nozzle is negligible compared to the cost of one which would be required for some particular liquid.

6 Finally Mr. Maker suggests the procedure to follow in determining relief areas, namely:

1 Determine the amount of heat input.
2 Determine maximum pressure of discharge.
3 Determine the vaporization rate.
4 Choose relief device to discharge vapor.

This is precisely the method used by the authors as shown in the paper.

Under item 1, we suggest 20,000 Btu per sq ft per hr, except for atmospheric tanks, as outlined in Part 3 of the paper.

Under item 2, we use the code rule in the paper of 10 per cent greater than the working pressure for maximum discharge pressure.

Under item 3, we determine the vaporization rate by use of the "contents factor."

Under item 4, we show a relation from which the minimum diameter of a circular relief area to discharge the vapor may be calculated.

We still feel that an "all-embracing" formula simplifies the problem for the host of equipment manufacturers and regulatory bodies who would need to go through these four steps, and that we are not trying to cover up any steps or introduce any irrelevant factors which would give unreasonable relief connections.

In answer to Mr. Maker's general conclusion, which of course refers to pressure vessels, the authors wish to point out that, whereas the information supplied by the paper may lack in application to large-size vessels, the existing available information is seriously lacking in application to small-size vessels.

Reply to Discussion by F. L. Newcomb

All of the actual fires used as a basis for the test of the curve described may not have been fires of great intensity. This committee should have asked for information from other industries besides the petroleum industry. A number of cases were cited by the authors in which the existing design rate of 6000 Btu was found to be inadequate, to which cases reference is suggested.

The only instance of a large tank being exposed, which supported the recommended curve, was that given by Turner C. Smith as example c-2. The apparent fallacy in its application is described under the reply to Mr. Smith's discussion to which attention is invited.

Reply to Discussion by R. C. Werner and S. T. Russell

The authors disagree with the tabulation of paragraph 6 and the statement of paragraph 7 in the comments by Mr. R. C. Werner and Mr. S. T. Russell, for the following reasons:

1 The very physical construction of the fuel around the propane cylinder would account for a variable heating rate on the wetted surface of the tank. As the wood is consumed and the fire bed lowers, the flame in the vicinity of the wetted surface would be more intense.

2 Up to the time of vaporization, the heat-transfer process is in an unsteady state. Thus, the steel vessel which weighs about 1000 lb and the propane vapor in the vapor space absorb heat, but as the temperature of the steel shell rises the rate of heat absorption decreases. However, as the fire level drops, then a relatively steady state is reached in the transfer of heat to the boiling liquid and practically all of the heat absorbed by the wetted surface goes into vaporization of the liquid. The values 1530 and 8410 Btu per hr per sq ft in the tabulation are evidently based upon the rise in liquid temperature only, and this low rate of sensible heat absorbed would be expected because of the mass of wood which surrounds the tank during the first few minutes of the fire.

Since these critics doubt the accuracy of Fetterly's test setup, as shown in Fig. 3 of the paper, the authors will be glad to display further information on request, including photographs and the results of two other tests conducted by Mr. Fetterly, or the critics may refer to files of the Bureau of Explosives in New York, N. Y.

In regard to the first point raised on the Underwriters' Laboratory tests, note that 461,000 Btu per hr (the heating rate at 1-min exposure) is still equivalent to 19,200 Btu per hr per sq ft. The chart shows that the plate heated from 65 F to 340 F in $^1/_2$ min, which is a heating rate of about $122 \times 0.12 \times 275 \times \frac{60}{0.5} =$ 483,000 Btu per hr. The maximum heating rate must be higher than this and exists at the first instant when the plate is at its lowest temperature. When the plate is at 600F (the temperature at 1-min exposure), it is losing heat from the rear and absorbs heat at a slower rate because of its high temperature.

The second point is rather abstract but actual observations of fires and the damage therefrom leave little doubt that air does exist in large areas so that combustion is complete enough to result in high temperatures. Reference may be made to the NFPA Handbook[9] on fire severity. Note also that a 3000-gal tank is not a small-scale tank but rather a commercial-process installation.

The third point is provided for in Part 3 of the paper. We agree that large-size tanks will not receive heat at the same rates as small-size tanks.

The comments on the Aluminum Company tests although lengthy are merely opinions of Messrs. Werner and Russell, and the authors of the paper do not grant that these opinions are superior to their own on these tests.

In regard to Items 4, 5, and 6, under the test made by the authors, note that the heat available in the fuel for test No. 4 is only 28 per cent of the quantity available in test No. 1. The two tests are, therefore, not comparable as far as determination of effect of water film is concerned. The emissivities for oxidized iron and steel, painted surfaces, and so on, are also reported as 0.95 to 0.98. The figure of 0.96 for water is calculated. Very few commercial surfaces will maintain a low emissivity for a long period of time.

In the summarization paragraph, Messrs. Werner and Russell refer to a large-area oil fire. We wish to point out that such a fire would not be as intense as a large-area butadiene or acetone fire. Large storage tank and large-area fire are not defined. The authors have given consideration to various sizes of tanks in using the observed heating rates.

It is the opinion of the authors that the summary statements of Messrs. Werner and Russell do not in any way detract from the observed heat-input rate of 20,000 Btu per hr per sq ft because:

1 They have calculated the input rate to a dry plate of 19,200 Btu per hr per sq ft (Underwriters' Laboratory tests); and if they are correct in their emissivities of iron and water, run No. 1 of the authors' tests would yield $25,900 \times 75/96 = 20,200$ Btu per hr per sq ft.

2 Damage to tanks and equipment in actual fires shows that vent sizes are too small whether they are venting air, vapor, or a combination of vapor and liquid.

3 A tank surrounded by other tanks and equipment would present the same windshield effect in an actual fire.

The authors would also point out that when human life is a consideration, it is customary to provide a factor of safety in the transposition of observed data to actual practice. For instance, in the design of pressure vessels a factor of safety of 4 or 5 is used. In view of the evidence of test data and the many cases of destruction of equipment in actual fires it is difficult to understand why anyone would desire to interpret data on the low side, or to design for any but the worst conditions that can be conceived.

Reply to Discussion by Walter Samans

The type of constructive criticism received from Mr. Samans is appreciated. The authors are in agreement with most of his comments. The following reply is made to several questions which he raised:

Mr. Samans feels that a heating rate of the order of one sixth of the observed rate (20,000 Btu) should apply to large atmospheric vessels. Solution of curve 3 in Fig. 18 of the paper will show that a rate, ranging from 5000 Btu at 300,000-gal capacity to 2300 Btu at 1,000,000-gal capacity, has been applied.

It was said that the authors' tests did not simulate fires in the open. This is true, as acknowledged, to an extent not great enough to affect materially the rate of heat absorption. The authors would point out that the test results were confirmed by five other independent organizations.

The reasons for stating, "So far as can be learned, there is no substantial theory or evidence to support the limiting of heat-input rates to any particular height. There are some indications to the contrary, that is, high gas temperatures apparently exist at relatively great heights above the fuel during large fires," will be found under the heading Flame Temperatures and Effective Height, near the end of Part 1 of the paper. In addition to the evidence there given, the authors have recently observed structural steel destroyed 30 to 40 ft above the liquid fuel. This was in the open and the structure was carrying only its own weight. The failures were the drooping type necessitating temperatures of at least 1000 to 1200 F.

Reply to Discussion by T. C. Smith

The following reply is made to specific quotations from Mr. Smith's discussion:

Mr. Smith's Comment. "It has been difficult to understand why the authors have chosen to determine the pipe size required for the relief-valve connection rather than the orifice area of the valve. Commercial spring-loaded relief valves have an orifice area ranging from 25 to 30 per cent of the connecting-pipe area, hence it would appear that the orifice should be of primary concern and the inlet-pipe connection of secondary interest."

Quotation Pertaining From Authors' Paper. "The relief capacity is proposed in terms of area of the relief connection (or connections) because the designer is primarily interested in selecting an adequate fitting of standard size for the vessel to be constructed, and because of the great variation in the capacity of commercial relief apparatus. This connection is ordinarily an inserted short tube with a square inner edge, somewhat rough from the standpoint of fluid flow. The relief device may afterwards be selected of a capacity (based on approved flow test) to pass the vapor rate of flow computed from relief area, as will be explained later."

Authors' Additional Comment. To elaborate further, the authors point out that of many popular types of safety valves studied, capacities were found to vary from 14 to 60 per cent of the capacities of tank connections of the same pipe size. It is generally recognized that only those capacities for valves which are essentially free-blowing orifices (top guided, high lift) can be calculated readily with practical accuracy. Such valves are in the minority of the many types manufactured. To repeat, the necessary relief capacity can always be calculated in terms of free area of the relief connection, and the apparatus chosen of a capacity (based on approved flow test) to pass the calculated rate.

The wide variation in the capacities of commercial relief devices and the questionable value of calculated capacities of valves which are not essentially free-blowing orifices will be obvious if reference is made to recent literature[45] on this matter.

It is further believed that the relief apparatus necessary for fire exposure, in most cases, would be supplemental to the usual spring-loaded safety devices and would also be a more simple, inexpensive device of relatively large capacity. Reference is suggested to paragraph 5, Proposed Summary for Use, Part 2, of the paper.

The method of specifying relief capacities in terms of free area is that shown in the "Proposed Tentative Standards: API Venting Guide" (May, 1940, p. 4). After Table 2, which gives the diameters of free areas with capacities, will be found the following statement in paragraph 7.1: "Certified capacity curves of venting equipment shall be based on flow tests conducted by commercial testing laboratories of institutions of recognized technical standing." Further, in the "A.S.M.E. Code for Power Boilers" (1940 edition, p. 108), will be found Table P-14 giving minimum total areas of openings (square inches) in fire-tube boilers for safety-valve connections.

The authors, therefore, feel that they have followed a recognized and rational method in expressing relief capacities in terms of free area.

Mr. Smith's Comment. "Also, consideration should be given to replotting the curves of 'contents factors' with the term 'pressure' incorporated in the equation. This will permit a series

[45] "Safety Valve Capacity Tests" (in accordance with the rules formulated by the A.S.M.E. Boiler Code Committee), National Board of Boiler and Pressure Vessel Inspectors, Columbus, Ohio, 1940; "Steam Flow Through Safety Valves," by E. K. Falls, Ohio State University, Columbus, Ohio, 1942; and "Safety and Relief Valves for Refrigerants," by E. K. Falls, *Refrigeration Engineering*, April, 1943, p. 257.

of charts, each consisting of a family of curves for various groups of contents."

Authors' Reply. This comment is not fully understood. It will be seen, in Fig. 12 of the paper, that the contents-factor values were plotted against pressure. No attempt was made to plot values for a substantial percentage of the numerous compounds but only for those considered representative and for which latent-heat data were directly available in the literature referred to in conjunction with Fig. 12. It is believed these are sufficient for the purpose of showing the method of solution and illustrating the value of simplification.

If a series of charts, consisting of values for families of compounds, were made, the present worth of Fig. 12 would be further enhanced. When trial solutions were made for numerous other compounds, the contents-factor values were found to be within the range of those now shown on the graph. "Their use in individual cases would result in numerous vent sizes adequate only for materials of like properties at the same condition." No different pipe-size relief connections other than those shown in Table 2 would result.

To emphasize again; the suggestion of a simplified value of the contents factor varying with pressure "in no degree nullifies the value of Equation [9] in calculating the required relief diameter directly for any contents."

Mr. Smith's Comment. "The authors present heat-input rates ranging from 20,000 to 30,000 Btu per hr as obtained from three different sets of tests."

Authors' Reply. Described and analyzed in Part 1 of the paper are tests by four (not three) separate organizations; seven tests in all. In the "Conclusion" of Part 2, further confirming tests by another company are referred to; while, in addition, four accidental fire exposures are pointed out as having supported the test data.

In the supplementary remarks to this paper made at Los Angeles on June 17, 1943, the authors cited one further test and one further actual example. The paper thus had for its practical basis the analysis and development of 10 man-made fire-exposure tests by six separate organizations and 5 actual cases confirming the test data. At this time, 6 additional failures due to fire exposure are being studied for further data.

Mr. Smith's Comment. "The speaker hastens to correct any doubts, on the deficiency of these venting rules, or impressions that this subject has been or is being neglected by groups engaged in the initial drafting of safety standards."

Authors' Reply. Early in Part 1, and again near the end of Part 2, the authors explained that the existing proposals for emergency venting were found to be inadequate. We believe this evidence to be convincing and will not repeat or elaborate on it here, but further explanation will be given in supplementary remarks. No inference was made that the subject was being neglected, but it was stated that the necessary information could not be obtained from the regulatory bodies. The following are quotations from a letter dated November 7, 1942, from the engineering office of one of the regulatory bodies in reply to our inquiry for information on the subject of rates of heat absorption during fire exposure:

The subject of emergency relief of internal pressure was headed up some twelve to fourteen years ago in the NFP A Committee on Flammable Liquids when the API Committee on Fire Prevention undertook to make studies and recommendations for inclusion in the NFPA Suggested Ordinance.

One of the first steps was a theoretical study along the lines of the enclosed, 'The Rates of Vaporization in Gasoline in Storage Tanks Exposed to Fire,' which was ultimately used as the basis for the typical bulk marketing tank of approximately 20,000 gallons capacity. There was a consensus among our fire prevention engineers that for such a tank we might reasonably expect absorption of heat energy at the rate of 100 Btu per minute per square foot of wetted surface.

More recently there have been fragmentary data which indicated much higher heating rates than 6,000 Btu per square foot per hour for the entire wetted surface of smaller tanks such as tank truck compartments. These data encouraged me to suggest a variable heating rate such as that represented by the red line on the chart. One oil company has recommended the use of this assumption in the form of the equation

$$h = \frac{48,000}{A^{0.8}}$$

where h = heat absorbed to cause evaporation—Btu per hour per sq ft.

A = external area of vessel.

We are lacking in data on larger tanks but there seems to be a general feeling among fire protection engineers that this assumption is reasonably safe, particularly for the larger sized tanks.

This letter is typical of the replies of other authorities on the subject, and this lack of data led to the independent study in an attempt to solve our own problem.

Mr. Smith's Comment. "The National Board of Fire Underwriter's rule was based upon a modification of Fetterly's formula, using a flame temperature of 1200 F for tanks having a capacity of 10,000 gal or less, and 1000 F for tanks of 30,000 gal capacity or more, with adjusted temperatures for tanks of intermediate capacity. The heat input by this rule will vary between 23,000 and 13,000 Btu per sq ft per hr."

Authors' Reply. If this reference (the last sentence quoted) is intended to apply to Fetterly's formula, it is not in agreement with the figures ("these values are not questioned") to be found under the heading Analysis of Fetterly's Formula and Test, in Part 1 of the paper, where Fetterly's work is analyzed and compared with present-day theory. There is shown the figure of 14,500 Btu per hr per sq ft of wetted surface for an effective flame temperature of 1200 F, and for 1000 F temperature, 8100 Btu will result.

Mr. Smith's Comment. The authors' paper was severely criticized for proposing a solution for large vessels without giving actual data on large-scale fire exposures. It should be noted that Mr. Smith cites only one large-scale analysis, which is as follows:

Example: "(c-2) Approximately 2300 Btu per hr per sq ft on a sphere having a wetted surface of 4363 sq ft. This was a 48-ft-diam sphere subjected to a severe fire for 1 hr 7 min. At times the vessel was completely enveloped in flame. The relief valves were adequate and prevented failure of the sphere."

Authors' Reply. The report of this same case, dated June 5, 1941, was received from one of the eastern refiners on November 6, 1942. In it will be found that the vessel was surrounded by fire for only 10 min. The result of 2300 Btu per hr per sq ft was the average for 1-hr exposure. Maximum rates of heat absorption may, therefore, have been 6 times as great (13,800 Btu) for which the relief devices should be adequate in order to limit the pressure rise. This is an excellent example but its maximum and not average values should be used as guidance toward safety.

As explained in Part 3 of the paper, it is expedient for practical reasons to lower the design heat-absorption rates for large atmospheric vessels. The vessel described contained 6000 bbl. When a solution is made of Equation [24] from curve 3, Fig. 18, of the paper, for a tank of this capacity, a heat-absorption rate of 3670 Btu per hr per sq ft of wetted surface results. This compares favorably with 2300 Btu average when it is considered that both are arbitrary assumptions, but the authors would use a much higher rate for the hazardous and expensive sphere under consideration in accordance with Part 2 of the paper.

For further evidence to support this reasoning, the authors have observed 17-ft-diam × 19-ft $4^1/_2$-in-high aluminum tanks

completely surrouned by fire and melted down in a matter of a few minutes. This metal had a melting point of 1216 F. An effective temperature of this magnitude is commensurate with a heat-absorption rate of 15,000 Btu per hr per sq ft. Actual effective hot gas temperatures were probably higher than 1216 F.

Reply to Discussion by H. R. Zeigler

In the paper we tried to avoid the discussion of relief devices, their relative merits, etc. Suggestions were, therefore, confined to relief areas or relief connections, and their calculated capacities. Mr. Zeigler's statements, "One novel suggestion is that relief valves be made the same size on all vessels of the same capacity and pressure," and ". . . it is necessary to point out that this is based on the assumption that tanks should be equipped with relieving devices of sufficient capacity to promote interchangeability," are misinterpretations. We did say: "It being proposed, as already inferred, that the relief connection be the same size on all vessels of given capacity and range of working pressure. When equipment is so provided with adequate relief area for volatile compounds in the form of relief connections, the relief devices may be replaced to provide for any change in contents or more severe working conditions from the standpoint of relief requirements." We feel that the relief apparatus should be selected to pass the calculated vent rate at the same conditions, and this choice will govern its size. This procedure is given in the section Proposed Summary for Use, of Part 2.

We were not aware that the National Board of Fire Underwriters' Pamphlet No. 58 applied to processing and storage equipment other than that for liquefied petroleum gases. We have used specifications for relief requirements from Pamphlet No. 30 but not from No. 58. For that reason we do not know and did not say that the relief areas in No. 58 are inadequate. We merely plotted specifications from the latter for comparison.

It is unfortunate that it appeared we were trying to prove the inadequacy of Fetterly's formula. As a matter of fact, the evidence indicates that his method has given more accurate results than any other. With a given latent heat (not calculated by $L = 0.185\ T\ (V1 - V2)\ dP/dT$ as he proposed) and for low pressures, our formula gives relief areas only 10 per cent less than Fetterly's. This comparison will be found plotted in Fig. 10. Fetterly's heat-absorption rate of 22,600 Btu per hr per sq ft is based on the vaporization of 1000 lb of propane taking place in the observed time of 5.9 min. The other figures are merely cited to support this rate, and it will be noted that there is reasonable agreement among them. This analysis was mailed to Fetterly for criticism but no reply was received.

Finally, the magnitude of the observed rate (20,000 Btu) of heat absorption is the principal contention. This is a long story about which we and our critics have written many pages. It may be pointed out that rates as high as 20,000 Btu have been confirmed by five other organizations besides ourselves. A number of critics are in agreement on this figure except for large vessels. A summary of the tests and experience on which the 20,000 Btu rate and 1400 F effective temperature are based has been published.[46] Further, it may be noted that the average rate for our four tests was 19,688 Btu per hr per sq ft at an average temperature for the flame of 1357 F (820 C being 1508 F).

We are confident that future tests and analyses of vessels exposed to accidental fire will prove the proposed heat rate to be of the correct magnitude.

[46] "Venting of Tanks Exposed to Fire," by J. J. Duggan, C. H. Gilmour, and P. F. Fisher, *National Fire Protection Association Quarterly*, October, 1943.

Wood-Cloth and Wood-Paper Laminates

By JOHN DELMONTE,[1] LOS ANGELES, CALIF.

A growing tendency exists for the use of wood and resin-impregnated cloth or paper, which is an outgrowth of laminated - plywood developments. Combinations of cloth and paper with wood, in addition to serving as vehicles for the bonding glue in the manufacture of the laminates, also help stabilize the wood assemblies and provide structural advantages, as demonstrated by test data given in the paper. For example, when a reasonably large number of paper laminates are used, there is an apparent increase in the tensile properties of wood, attributable to prestressing of the wood resulting from differences in coefficients of thermal contraction as parts are cooled upon removal from the press. Dimensional stability in plywood faced with phenolic-resin-treated cloth and paper is found to be far superior to unprotected plywood.

A NATURAL consequence of laminated-plywood developments and laminated-plastic-material developments is a trend toward intermediate products combining wood and resin-impregnated cloth or paper. While it will be acknowledged that there are applications best served by wood alone or other applications best fulfilled by plastics materials, there are various properties developed by their combination that are not possessed fully by either material itself. There are not many evidences as yet of extensive adaptation of resin-impregnated cloths or papers with wood although a few are in the process of development for aircraft applications.

Examining the fundamental properties of wood, one will admit its superior advantages of low cost, light weight, and high strength although he will also recognize its anisotropic tendencies and susceptibility to moisture. Plywood developments were a logical evolution in that they balanced physical properties more evenly and reduced the effects of swelling. However, while resin-bonded plywoods were a step in the right direction, even these have not always possessed the necessary dimensional stability required of aircraft materials, in spite of synthetic-resin bonding agents, such as phenol formaldehyde, urea formaldehyde, and melamine formaldehyde. Elaborate finishing schedules offer some measure of protection to the plywood, but even these are beset by numerous practical problems (1).[2]

The most noteworthy developments for stabilizing plywood have been suggested by the Forest Products Laboratory (2, 3) in their publications upon resin-impregnated wood. This well-known technique involves a procedure of impregnating thin wood veneers with a water-soluble phenolic resin and curing the resin in the veneer by heat alone (Impreg), or by combination of heat and pressure (Compreg). Approximately 30 per cent resin diffused into the cell walls of the wood fibers has resulted in a decided improvement in wood-stabilizing characteristics, particularly against swelling effects of moisture. What few objections have been raised against these techniques include the following:

1 Increase in density without a substantial gain in strength.

2 Added difficulties in performing secondary gluing operations in aircraft as compared with unimpregnated plywood.

3 Problems of obtaining proper impregnation and its determination.

Also to be noted are the developments of superpressed plywood and multiple-phenolic-resin films which increase the ratio of resin to wood (4). High laminating pressures above 1000 psi have been employed to increase the penetration of resin into the wood. The cardinal advantage of this process is the development of superior shear strength in the plywood, which greatly increases for a slight increase in density. On the other hand, the rate of increase of tensile strength and compressive strength over density is only slight. This development is also significant in the technique of increasing resin content by adding multiple layers of phenolic-resin film, a procedure which would be unnecessarily prolonged if spraying were resorted to.

Both of these fairly recent developments in stabilizing plywood have taken into account the advantages of increasing the ratio of synthetic resin to wood in order to stabilize the wood. Ordinary synthetic-resin-bonded aircraft plywood contains anywhere from 10 to 25 lb per 1000 sq ft of dry-glue spread sufficient for good bonding, high shear strength, and good water resistance at the glue lines only.

The combinations of resin-impregnated cloth and papers with wood fulfill more than the function of vehicles for the glue; they also help to stabilize the wood assembly as well as make possible structural advantages as various test data disclosed in this paper will reveal. In addition, the wood-cloth and wood-paper laminates make possible more complicated assemblies and constructions, difficult with plywood alone.

Physical Test Conditions

Unless stated otherwise the various physical tests reported upon in this paper were performed upon wood-cloth or wood-paper laminates which had been conditioned at 50 per cent relative humidity for at least 48 hours in order to realize approximately 10 per cent moisture in the wood. Tests were performed under prevailing atmospheric conditions which, for the most part, were at room temperatures of 70 to 80 F, and relative humidities of 35 to 45 per cent. All tensile tests and flexural tests followed A.S.T.M. Standards for flat sheets of plastics materials (5). Specific-gravity determinations were made by measuring the volume of accurately machined samples and dividing this value into the weight. The moduli of elasticity reported were obtained by the simple cantilever-beam technique. This method is described elsewhere in its adaptation to plastics materials (6).

Test Laminates

The following laminates were prepared for physical tests and tests of dimensional stability under extremes of moisture condition. In Groups I and II the base material was standard-aircraft-grade 3-ply vertical-grain plywood with two birch faces and poplar core.

Group I:

(*A*) Plywood base, two birch faces, poplar core; over-all thickness 0.106 in.

(*B*) Resin-impregnated paper,[a] 2 layers on each face of plywood-base core (*A*)

(*C*) Resin-impregnated paper,[a] 5 layers on each face of plywood-base core (*A*)

[1] Technical Director, Plastics Industries Technical Institute. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at end of paper.

Contributed jointly by the Rubber and Plastics Group and Aviation Division and presented at the Semi-Annual Meeting, Los Angeles, Calif., June 14–17, 1943, of The American Society of Mechanical Engineers.

(*D*) Resin-impregnated paper,[a] 10 layers on each face of plywood-base core (*A*)

[a] Consolidated Water Power & Paper Company; phenolic-resin content, 35 per cent.

NOTE: Group I samples were laminated at 250 psi for 10, 15, and 20 min at 310 to 320 F, for increasing layers of paper.

Group II:

(*A*) Plywood base, two birch faces, poplar core; over-all thickness 0.106 in.
(*B*) Phenolic-resin-impregnated canvas,[b] 1 layer on either face of plywood-base core (*A*)
(*C*) Phenolic-resin-impregnated canvas,[b] 2 layers on either face of plywood-base core (*A*)
(*D*) Phenolic-resin-impregnated canvas,[b] 5 layers on either face of plywood-base core (*A*)

[b] 9-oz canvas duck, phenol-formaldehyde impregnating resin varnish (BV-1115), 38 per cent resin content.

NOTE: Group II samples were laminated at 250 psi for 20, 25, and 30 min at 310 to 320 F for increasing layers of cloth.

Group III:

(*A*) Three-ply birch veneer, 0.032 in. thick, core cross-grain; phenolic-resin bonding agent, PR-14 (Amberlite)
(*B*) Similar to (*A*) except 2 phenolic-resin-impregnated papers added between each ply
(*C*) Similar to (*A*) except 5 phenolic-resin-impregnated papers added between each ply

NOTE: Group III samples were laminated at 250 psi, and 325 F for 15 min. Plywood shear tests were performed in accordance with AN-NN-P-511b Army-Navy Specification for Plywood and Veneer, dated October 28, 1942.

Group IV:

Two parallel-grained birch veneers, $^1/_{16}$ in. thick, between which are laminated:

(*A*) 1 phenolic-resin-impregnated paper
(*B*) 5 phenolic-resin-impregnated papers
(*C*) 10 phenolic-resin-impregnated papers
(*D*) 20 phenolic-resin-impregnated papers

Group V:

Two cross-grained birch veneers, $^1/_{16}$ in. thick, between which are laminated:

(*A*) 1 phenolic-resin-impregnated paper
(*B*) 5 phenolic-resin-impregnated papers
(*C*) 10 phenolic-resin-impregnated papers
(*D*) 20 phenolic-resin-impregnated papers

NOTE: Groups IV and V were laminated at 250 psi and temperature of 320 F for 20 min (*A* and *B*), 25 min (*C*), and 30 min (*D*).

RESULTS OF TESTS ON PHYSICAL PROPERTIES

The results of tests on the laminated assemblies may be divided into two groups, i.e., physical properties and dimensional stability under extremes of moisture conditions. The physical properties are presented first.

Results of physical tests upon Group I and Group II laminates are shown in Table 1. In order to keep variables to a minimum, laminating pressures of 250 psi are employed for these laminates as well as the others. This pressure likewise is not too far removed from those employed in rubber-bag-molding operations. Data reported for the straight plywood (10 per cent moisture) compare favorably with those reported in other publications (7). The most obvious improvement in physical properties is apparent in the wood-paper laminates (Group I). Not only are values of the tensile strength and modulus of elasticity decidedly improved but also the specific tensile strength, which is equal to the ultimate tensile value divided by the specific gravity. These improvements are due largely to the excellent properties of the phenolic-resin-impregnated paper employed in conjunction with the plywood.

Manufacturers' specifications for laminates of paper alone at 250 psi are as follows:

Ultimate tensile strength (with grain) 36,000 psi
Modulus of elasticity (with grain) 3,000,000 psi
Specific gravity, 1.38
Water absorption, 6 per cent (24 hr)

Unless indicated otherwise the test data of this report were obtained in a direction parallel to the grain of the impregnated paper. It was further observed that properties for the straight laminated paper decreased somewhat (tensile strength at 33,000 psi, for example), when the material was conditioned for several days at 50 per cent relative humidity. It was pointed out earlier that all the wood laminates were conditioned in this manner.

Table 1 also reports the stiffness factors ($E \times I$) for the various laminates assuming, for purposes of comparison, a constant panel weight of 0.5 lb per sq ft and then estimating moments of inertia from

$$I = \frac{bh^3}{12}$$

where b = 1 in.
h = thickness which at specific gravity indicated in Table 1 will give 0.5 lb per sq ft
I = moment of inertia.

Plywood faced with a few sheets of impregnated paper shows slight gains in the stiffness factor. While plywood faced with resin-impregnated canvas does not register the gains shown for the wood-paper laminates, the data nevertheless demonstrate the versatility of plastic materials in securing moduli of elasticity at very much lower values than with plywood alone. To the stress engineer this suggests decreased stiffness factors, but to the fabricator of compound curvatures it suggests an easier material to fabricate plus the added advantage of better dimensional stability under adverse weather conditions.

The load distribution within the various components of wood-cloth and wood-paper laminates are basically dependent upon the moduli of elasticity of the different components. For pure tension and pure compression it is obvious that

$$\delta = \frac{S_1}{E_1} = \frac{S_2}{E_2}$$

TABLE 1 PHYSICAL PROPERTIES OF SOME CLOTH-WOOD AND PAPER-WOOD LAMINATES[a]

Laminated assembly	*A* Specific gravity	*B* Ultimate tensile strength, psi	*B/A* Specific tensile strength, psi	Modulus of elasticity, psi	Stiffness factor,[b] $E \times I$	Ratio of resin to inert material,[c] per cent
Laminated phenolic (canvas)	1.33	17500	13200	1080000	34	38
Plywood (birch faces, poplar core)	0.74	13680	18500	1780000	326	0
Plywood, faced with 2 layers phenolic-resin-treated paper	0.76	13250	17500	2250000	381	4.1
Plywood, faced with 5 layers phenolic-resin-treated paper	0.86	15270	17700	2250000	264	8.1
Plywood, faced with 10 layers phenolic-resin-treated paper	0.93	19000	20400	2320000	211	15.2
Plywood, faced with 1 layer phenolic-resin-treated canvas	0.82	11800	14400	1225000	163	6.3
Plywood, faced with 2 layers phenolic-resin-treated canvas	0.92	11600	12600	1228000	115	15.6
Plywood, faced with 5 layers phenolic-resin-treated canvas	1.08	11500	10600	1040000	61	24.7

[a] Prepared at 250 psi.
[b] Based on weight of 0.5 lb per sq ft.
[c] 35 per cent phenolic resin in paper; 38 per cent phenolic resin in canvas.

TABLE 2 RESULTS OF SHEAR TESTS ON PLYWOOD[a]

Plywood combination	Dry shear tests (average of 5 tests), psi	Results
3-ply birch, resin alone as bond	637	95–100 Per cent wood failure
3-ply birch, resin plus two layers of impregnated paper in each glue line	577	Approximately 70 per cent failure in paper
3-ply birch, resin plus five layers of impregnated paper in each glue line	542	Approximately 70 per cent failure in paper
Specification allowance	400	

[a] 3-ply, all birch, $^1/_{32}$-in-thick veneers.
NOTE: Adhesive, phenol-formaldehyde resin (Amberlite PR-14). Additions to adhesive, phenolic-resin-impregnated paper (Consolidated Water Power & Paper Company). Test specification, AN-NN-P-511b (Army-Navy Aeronautical Specification Plywood and Veneer).

TABLE 3 PHYSICAL PROPERTIES OF GROUPS IV AND V LAMINATES

Laminate	Ultimate tensile strength, Psi	Modulus of elasticity, Psi
Parallel-grain birch:		
Faces 1, impreg; paper core	20100	1600000
Faces 5, impreg; paper core	22970	1850000
Faces 10, impreg; paper core	24490	2050000
Faces 20, impreg; paper core	29400	2400000
Cross-grain birch:		
Faces 1, impreg; paper core	3220	137000
Faces 5, impreg; paper core	7640	170000
Faces 10, impreg; paper core	12200	250000
Faces 20, impreg; paper core	17670	491000

where δ = unit deformation in composite assembly of material No. 1 and material No. 2
S_1 and S_2 = unit stresses in No. 1 and No. 2
E_1 and E_2 = moduli of elasticity in No. 1 and No. 2

Likewise, for calculating stress at any point within the plywood panel, the formula developed for straight plywoods (8) may be applied to cloth-wood and paper-wood laminates, considering the layers of paper or cloth in the same category as a layer of wood veneer

$$S_L = \frac{E_L}{E_P} \cdot \frac{M_P C}{I_P}$$

where S_L = stress in layer or laminate in question
E_L = modulus of elasticity of that laminate alone
M_P = bending moment at point in question
C = distance from neutral axis to layer or laminate in question
E_P = apparent modulus of entire plywood panel (see Table 1)
I_P = moment of inertia of entire plywood panel

Group III laminates were prepared largely with the idea of observing the effects of impregnated paper upon the shear strength of plywood. Samples were cut in accordance with the Army-Navy Aeronautical Specification for plywood and veneer (9) and tests performed as required. Results of these tests, disclosed in Table 2, demonstrate that all the plywood samples with and without impregnated papers at the glue line, passed the minimum requirements of the specification. However, there was a small, but definite decrease, in shear strength with an increase in paper content. This was occasioned by the appearance of paper failure which was marked by a separation of the fibers of the paper. Dry shear tests were performed upon five samples for each reading record in Table 2, after 48 hr conditioning at 50 per cent relative humidity.

Groups IV and V laminates serve the purpose of revealing further fundamental data on the co-relationship of paper and wood. While the plywood-paper samples prepared are unbalanced in the sense of uniform grain direction for the wood, without the benefit of cross-plies, results on the load distribution within the wood and impregnated paper are interesting. In Fig. 1, the marked increase in tensile strength, for example, is to be expected from the excellent properties of the paper employed. As indicated earlier, the ultimate tensile strength of this particular laminated paper alone is 36,000 psi, after dry conditioning, and about 33,000 psi after 50 per cent relative-humidity conditioning. It is significant, however, that the increase in tensile strength of the wood is greater than would be expected upon the addition of the paper. This is illustrated on the right-hand side of Fig. 1. In other words, assuming constant tensile strength for the laminated paper, the apparent strength of the wood increases with greater amounts of impregnated paper. These data were obtained, for pure tension, as follows:

$$P_1 + P_2 = P = A_1S_1 + A_2S_2$$

where P_1 = portion of load carried by impregnated paper
P_2 = portion of load carried by wood veneer
A_1 = cross-sectional area of paper laminate
A_2 = cross-sectional area of wood veneer
S_1 = stress intensity in paper laminate
S_2 = stress intensity in wood veneer
P = total load

Using the foregoing formula and calculating load distribution for increasing amounts of paper laminate, the increase in P could not be accounted for solely by the increase in high-strength-paper content. However, it is possible that thermal-expansion coefficients vary and upon removal of the laminate from the hot plates of the press, the wood veneers were under initial stress due to greater shrinkage of the laminated phenolic-paper layer upon cooling. Then

$$P = A_1S_1 + A_2S_2 + KE_2A_2$$

where K = unit deformation due to thermal-expansion difference during manufacture
E_2 = modulus of elasticity of wood veneer

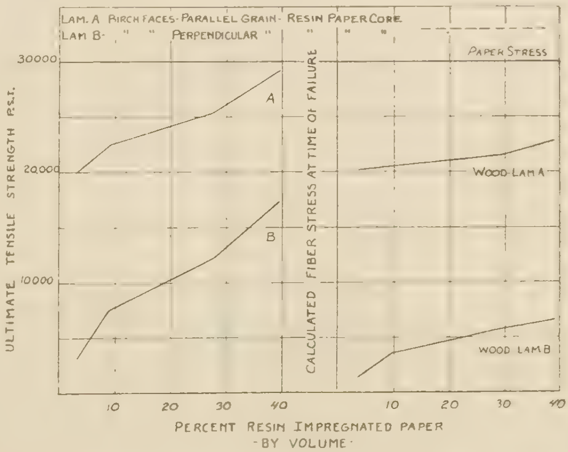


FIG. 1 ULTIMATE TENSILE STRENGTH OF WOOD-PAPER LAMINATES

In Table 3, are reported physical properties of Groups IV and V laminates after conditioning at 50 per cent relative humidity.

Results of Tests on Dimensional Stability

The effects of moisture gradients upon wood laminates are to bring about severe dimensional distortion. While the developments of synthetic-resin adhesives for the glue lines have created the so-called "waterproof" plywood, in the sense that shear failures would no longer be due to deterioration of the glue line the resin adhesives will not protect the wood structure itself. Faced with very large tangential and radial shrinkage in all wood veneers, plywood manufacturers are ever concerned with the avoidance of warping and distortion in plywood panels. Much care is necessary in preconditioning the wood veneers to uniform moisture content, and then in afterconditioning to compensate for loss of moisture in hot-pressing or the introduction of moisture from the glue.

As pointed out earlier, synthetic-resin impregnants have stabilized wood by greatly reducing the effects of moisture. Some stabilization is achieved by facings of impregnated canvas and impregnated paper on the outside of the plywood core (Groups I and II laminates) by creating a moisture barrier. The effectiveness of phenolic-resin-impregnated paper and cloth in stabilizing plywood may be determined by means of a test designed to measure distortion in plywood upon exposure to severe moisture gradients.

As shown in Fig. 2, one side of a plywood test sample was coated with a metal face, 1 mil in thickness, which extended up the sides of the test sample, excluding moisture from all surfaces except the top surface. Upon placing the plywood sample in a pan of water, immediate deformations are apparent as the plywood fibers in direct contact with the water begin to swell. This swelling or distortion in the 5-in. test length is measured with a depth gage, as shown in Fig. 2. Results are clear-cut and comparisons may be made in a short period of time, without any delay being necessitated for the less-significant percentage of water absorbed.

All test samples were conditioned at 50 per cent relative humidity for at least 48 hours prior to test, and distortion is less than would have occurred if the materials were started from a perfectly dry condition. However, the interest lies largely in the comparative results.

Data shown in Figs. 3 and 4 illustrate the distortion in plywood as a function of time. It is quite obvious that resin-treated papers and cloth greatly improve the dimensional stability of plywood by retarding the effects of moisture. The distortion curve for plywood alone demonstrates a peak value, indicative that moisture has diffused through the plywood and that the fibers nearest the protective metal foil have begun to swell counteracting the swelling on the exposed side. When the test is carried out for a long period of time the distortion in the plywood is largely recovered.

For purposes of comparison, the more conventional A.S.T.M.

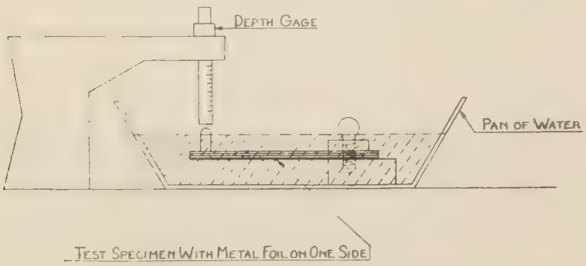


Fig. 2 Test for Dimensional Stability

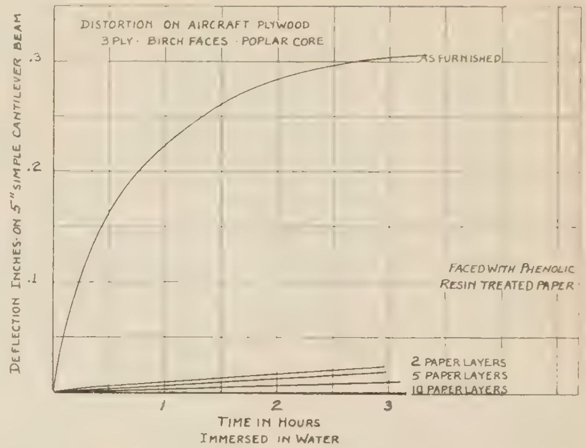


Fig. 3 Distortion in Plywood Faced With Phenolic-Resin-Treated Paper

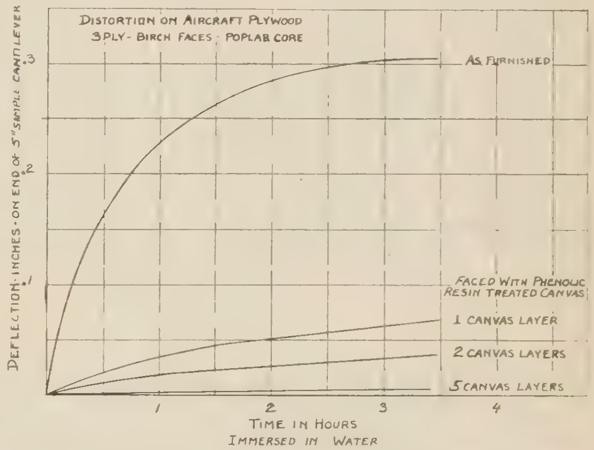


Fig. 4 Distortion in Plywood Faced With Phenolic-Resin-Treated Canvas

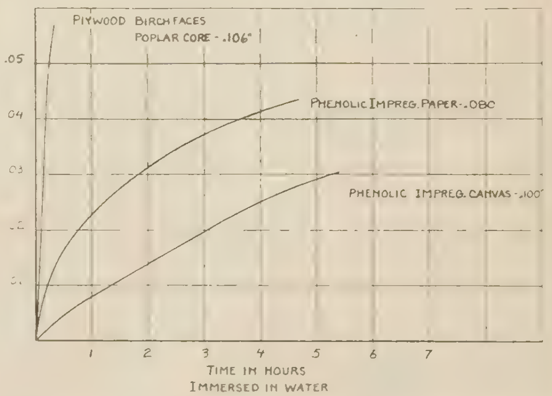


Fig. 5 Distortion of Laminated Canvas and Laminated Paper in Contact With Water

tests were performed upon plywood faced with one and two layers of phenolic-resin-treated canvas. Pieces which were dry-conditioned measured 1 in. × 3 in., the sides being protected against moisture by metal foils. The results follow:

Material	Water absorbed (in 24 hr), per cent
Plywood faced with one phenolic-resin-treated canvas	8.4
Plywood-faced with two phenolic-resin-treated canvas	3.1

As may be suspected, the plywood distortion in moisture may not be attributed to the plywood alone, but also to the resin-impregnated canvas and paper. This is borne out by the disclosures in Fig. 5 which show distortion of laminated canvas and laminated paper when they are brought in contact with water as shown in Fig. 2.

That some of the combinations with plywood show even lower distortions is explained by the greater over-all thicknesses of these assemblies.

Conclusions

It is apparent that there are physical advantages to be realized from the combinations of resin-impregnated paper and cloth with plywood; particularly for the high-strength paper employed in these tests. There is some evidence that when a reasonably large number of paper laminates are present there is an apparent increase in the tensile properties of the wood attributable to prestressing of the wood brought on by differences in coefficients of thermal contraction as parts are cooled upon removal from the press.

While resin-impregnated canvas did not add greatly to the physical properties of the plywood, the molders and fabricators of plywood parts would do well to recognize the benefits of canvas in its stretchability, allowing the attainment of smaller radii of curvature and a certain amount of drawing impractical in wood alone. This would suggest the usefulness of phenolic-resin-impregnated canvas in fulfilling discontinuities in a cloth-wood combination.

Dimensional stability in plywood faced with phenolic-resin treated cloth and paper is far superior to unprotected plywood. This advantage alone should suggest to manufacturers of aircraft plywood that a few facings of impregnated paper or cloth would improve the qualities of their products.

Acknowledgment

Acknowledgment is given to Mr. Foster Luce, Mr. E. Watkins, and Mr. W. F. Lusk of the Plastics Industries Technical Institute staff for assistance in preparation of some of the samples and performance of some of the physical tests, data for which are given in this paper.

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Superchargers for Aircraft Engines

By R. G. STANDERWICK[1] and W. J. KING,[1] WEST LYNN, MASS.

This is a comprehensive treatment covering the development of turbosuperchargers, which today are making possible the outstanding performance at high altitudes of United States military aircraft. The basic function of supercharging is to increase the intake-manifold air pressure in an engine, as a result of which the primary objectives are attained of (*a*) maintaining full (rated) power at altitude; (*b*) increasing power at sea level; (*c*) improving fuel economy for cruising. Fundamental design considerations cover the means for accomplishing these ends, with some indication of the power requirements, temperature effects, and the like. Types of superchargers are discussed, and their systems of drives, and then particular attention is devoted to the General Electric turbosupercharger, as used on the Boeing Flying Fortress, Consolidated Liberator bomber, and Lockheed Lightning, and Republic Thunderbolt pursuit ships. Final sections of the paper are concerned with a comparison of various types of superchargers, and the present and future prospects for this most important aircraft-engine auxiliary.

Fundamentals of Supercharging

The basic function of supercharging is to increase the intake-manifold air pressure in an engine. This has two immediate results; (*a*) the density of the charge forced into each cylinder is increased, and (*b*) a favorable excess or differential is maintained in the intake manifold pressure with respect to the exhaust back pressure. These two effects are utilized to accomplish the primary objectives of supercharging aircraft engines, as follows:

1 To maintain full (rated) power at altitude.
2 To increase power at sea level ("ground boosting").
3 To improve fuel economy for cruising.

A further effect of supercharging, which is beneficial in each of these respects, is to promote fuel vaporization and uniform distribution of mixture to the individual cylinders.

In order to appreciate the nature of the problem, in reference to these objectives, it is first necessary to take account of a few of the elementary relationships underlying the performance of present-day aircraft engines.

Other things being constant, the indicated horsepower developed in an engine is directly proportional to the mass rate of air consumption (as pounds per minute). For a given speed this means that the power is proportional to the density of the air, which is a function of its pressure and temperature. As a fairly close approximation (1),[2] the brake horsepower of an engine varies directly with the absolute pressure, p, and inversely as the square root of the absolute temperature, T, of the carburetor inlet air, as expressed in the formula

$$\frac{\text{Bhp}_0}{\text{Bhp}_1} = \frac{p_0\sqrt{T_1}}{p_1\sqrt{T_0}} \quad [1]$$

where the subscripts 0 and 1 refer to standard sea-level and altitude conditions, respectively.

Altitude Effects. Fig. 1 shows how these factors vary with altitude in a representative modern aircraft engine, except unsupercharged, using standard N.A.C.A. atmospheric data (2). A glance at this should suffice to indicate the occasion for supercharging military-airplane engines, where it is so vitally important for the engine to be capable of developing its full power at any level within its altitude range.

The significant fact here is that by the application of a suitable supercharger the same engine which gave less than 30 per cent

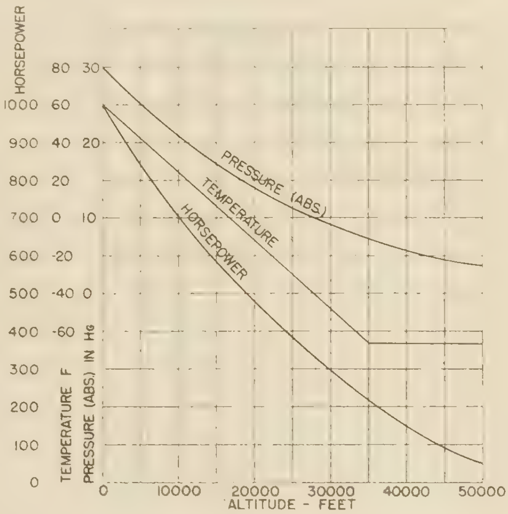

Fig. 1 Variation in Atmospheric Pressure and Temperature With Altitude and Effect Upon Engine Power

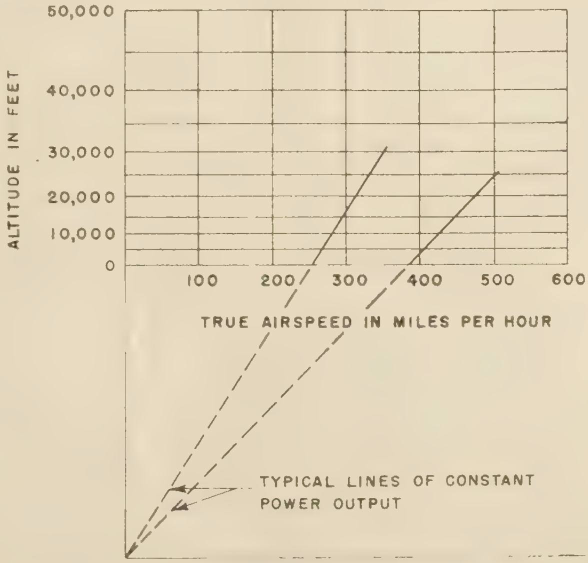

Fig. 2 Airplane Speed With Constant Horsepower Versus Altitude

[1] Supercharger Engineering Division, General Electric Company.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Aviation Division and presented at the Semi-Annual Meeting, Los Angeles, Calif., June 14–17, 1943, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

of its full power at 30,000 ft can be made to deliver 100 per cent power all the way from sea level to that altitude, and beyond. With proper attention to such factors as propeller design, propeller and engine speeds, and type of supercharger, it is further possible to maintain substantially the same fuel-consumption rate (pounds per horsepower-hour) at altitude as at sea level.

These two facts have a direct bearing upon the curves in Fig. 2, which shows the variation in the speed of a given airplane with altitude, if the engine power output can be held constant. This chart is calculated from theory and assumes constant propeller and airplane efficiency. Under these conditions, it may be seen that a plane having a speed of 250 mph at sea level may attain a speed of 350 mph at 30,000 ft, for the same power and time rate of fuel consumption. Of course actual airplanes will not follow this chart exactly, but Kendall Perkins of Curtiss Wright states that, on the average, the speed of a plane will increase by 1 per cent for each 1000 ft of altitude (3). Very likely a higher figure can be realized in a suitably designed plane. At all events it is clear that the ability to maintain power at high altitudes results in marked improvements in peak performance of the plane, saving valuable time in military or commercial operations, and increasing the speed at which satisfactory over-all economy can be maintained.

Ground Boosting. Superchargers can also be used very effectively for ground boosting, i.e., increasing the sea-level power of the engine, since the indicated mean effective pressure (imep) and horsepower are proportional to the intake-manifold air pressure (map). With a moderate allowance for friction and accessory power loss in the engine, the brake-mean-effective-pressure values, (bmep) and brake horsepower (bhp) likewise increase very nearly linearly with the manifold air pressure. Of course, the power output of an engine can also be increased by increasing the compression ratio, but the interesting fact here is that a given increase in bmep and bhp can be obtained with a considerably smaller rise in maximum cycle pressure and temperature by means of supercharging. The result is that the manifold pressure can be boosted to yield a substantially greater power output before the allowable limit is imposed by stresses, temperatures, or detonation. It is entirely practicable, for example, to increase the output of an unsupercharged engine from 500 to 800 hp by boosting the manifold air pressure from 29 to 40 in. Hg abs, without changing the speed. This extra power is frequently of great value in facilitating take-off.

In all of the foregoing comparisons, the engine speed has been assumed to remain constant. The variation of power with speed is expressed by the formula

$$\text{Bhp} = \text{const} \times \text{bmep} \times \text{rpm} \qquad [2]$$

High power is therefore commonly obtained by a combination of high speed and high bmep. High speed is attained by setting the variable-pitch-propeller governor to hold the desired revolutions by appropriate adjustment of the propeller-blade angle, and high bmep is secured by boosting the manifold pressure either by opening the throttle or increasing the speed of the supercharger (if the latter is controllable).

Cruising Power. For cruising, the power can be reduced either by reducing the speed or the bmep, or both. If the propeller efficiency is not impaired, it is definitely more economical to reduce power by reducing the speed, keeping the bmep as high as practicable. This is a consequence of the reduction in engine friction losses at the lower speeds. When the speed is reduced for this purpose at cruising altitudes, it is frequently found that the most economical values of map and bmep cannot be attained even with full opening of the throttle without a considerable amount of boosting from the supercharger. As an example to help make this clear, the following figures are cited for a hypothetical engine for which the value of the constant in Equation [2] is 0.0025:

Power	Cruising (maximum economy)	Full
Manifold air pressure, in. Hg.....	30	47
Brake mean effective pressure, psi.	140	220
Speed, rpm.....................	1200	2700
Power, bhp.....................	420	1485

At 15,000 ft, the manifold pressure could not be more than about 16.5 in. Hg without supercharging. This would yield a bmep of approximately 77 in. Hg, which would require a speed of 2180 rpm to produce the 420 bhp for cruising. The fuel consumption at this speed would normally be appreciably greater than at 1200 rpm. To maintain 420 bhp at still lower speeds would probably require richening of the fuel-air mixture to avoid detonation, and the propeller efficiency might be considerably reduced.

Supercharger Operating Range. Another serious obstacle to the maintenance of high manifold pressures for cruising is the limited range of satisfactory operation of superchargers now in common use, as indicated in Fig. 3. In this figure, Q represents

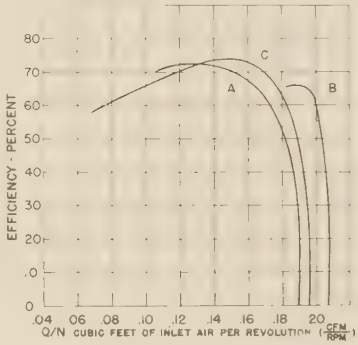

Fig. 3 Typical Range Curves of Superchargers

the volume flow of air supplied to the engine cfm, and is proportional to the engine power, at any given altitude; N is the supercharger speed, rpm, which is the chief factor in determining the pressure (map) supplied to the intake manifold. The useful operating range of a supercharger refers to the spread between the lower value of Q/N, usually established by the incidence of violent surging or instability, and the upper value beyond which the efficiency falls off very rapidly. Curve A is representative of the modern centrifugal supercharger. The significance of these curves will be discussed further. For the present purpose it will suffice to observe that curve A indicates the difficulty of attempting to reduce Q for very low power outputs while maintaining N at a relatively high level, in order to employ high manifold pressures. In addition to the fact that this tends to push the value of Q/N into the unstable region beyond the left end of the curve, a very flexible speed control is required to allow N to be maintained when the engine speed is reduced to less than one half of its maximum rating.

Differential-Pressure Effects in the Engine. Apart from its prime function of increasing the density of the charging air forced into the engine cylinders, supercharging has a further beneficial effect in promoting the scavenging of spent gases from the clearance volume, at the end of the exhaust stroke. Since the intake-manifold pressure is higher than the back pressure some of the hot residual gas is forced out during the "overlap" interval between the opening of the intake valve and the closing of the exhaust. This improves the volumetric efficiency, lowers the initial temperature of compression, and reduces the tendency to backfire with lean fuel-air mixtures.

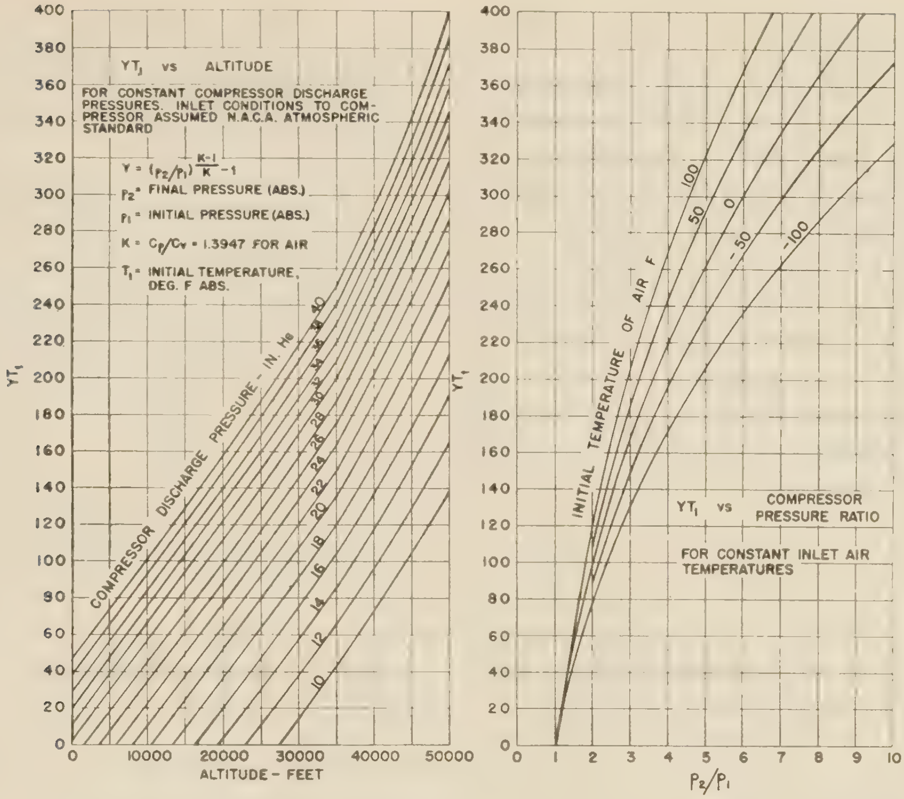

Fig. 4 Variation of Function YT_1 With Altitude, Pressure Ratio, and Initial Temperature

The excess of the intake pressure over the exhaust must be taken into account in computing the effect upon the net engine power of the power required to drive the supercharger. A substantial portion of the latter, roughly 40 per cent, is returned to the engine in the form of "steam-engine power," which is the excess of the work done on the pistons during the intake stroke over the work done on the spent gas during the exhaust stroke.

Power Required for Driving Supercharger. Present-day aircraft engines consume about 0.12 lb of air per min per hp delivered. The power required to compress this air is determined chiefly by the ratio of the final to the initial absolute pressures, p_2/p_1, and the absolute temperature, T_1, of the inlet air. The theoretical power for isentropic compression is given by the formula (reference 4)

$$\text{Hp} = 0.00573\ YT_1 \text{ per lb air per min} \qquad [3]$$

where

$$Y = (p_2/p_1)^{0.283} - 1$$

and T_1 is in deg F abs.

Values of the function YT_1 are given in Fig. 4. The curves on the left are based on the assumption that p_1 and T_1 correspond to the standard N.A.C.A. atmospheric values of Fig. 1, for each altitude. The curves on the right give the same function for the particular pressure ratio and inlet temperature involved in any case. To obtain the compressor-shaft-input power, the values obtained from Equation [3] must be divided by the efficiency.

In many practical cases p_1 may be assumed to be equal to the prevailing atmospheric pressure in flight, since the pressure drop in the upstream induction system is roughly balanced by the "ram" or kinetic pressure of the slipstream impinging upon the air scoop. For the typical internal-geared supercharger p_2 is the manifold pressure. Assuming a compressor efficiency of 70 per cent, the power required to develop 1000 hp at 10,000 ft with a manifold pressure of 32 in. Hg can then be obtained from Fig. 4 and Equation [3] as follows

$$YT_1 = 65$$

$$\text{Hp} = \frac{0.00573 \times 65 \times 120}{0.7} = 63.8$$

(This must again be divided by the mechanical efficiency of the drive train to obtain the power taken from the engine crankshaft.) The pressure ratio in this case is 1.55. For comparison, a supercharger to maintain 40 in. Hg of manifold pressure at 25,000 ft would have to develop a pressure ratio of 3.6, which would require an input of about 185 hp per 1000 hp of engine power.

The newcomer to this field sometimes finds it difficult to understand how a device which extracts such sizable blocks of power from the engine can be applied to sustain, or even increase, the net engine output. Perhaps this can be appreciated more readily from the analogy which it bears to the forced-draft blower of a stoker-fired furnace of a steam power plant, where a few hundred horsepower input to the blower enables the furnace to burn enough additional coal to generate several times the initial investment in power. Moreover, as mentioned, the supercharger compressor actually pumps a substantial portion of its shaft power back into the engine (5). For the electrical engineer the supercharger may be compared with the grid of an electronic tube, when the grid system is used to control the relatively large plate-circuit power in a regenerative hookup. The fact is that the supercharger is being employed to an increasing degree as the

primary means of controlling the power of an airplane engine, in place of the carburetor throttle.

Temperature Effects. Unfortunately, air cannot be compressed even with the perfect efficiency of an isentropic process, without a significant rise in temperature, unless the heat of compression is removed by cooling during the process. In aircraft superchargers it is not practicable to effect any appreciable amount of cooling in the compressor itself; in fact, the surface area is so small relative to the mass flow of air that the process is substantially adiabatic (and polytropic).

For an adiabatic process, the minimum temperature rise occurs in the case of reversible, isentropic compression, i.e., when the compressor efficiency is 100 per cent. In this case, the ideal temperature rise is numerically equal to the function YT_1 of Fig. 4

$$T_2 - T_1 = YT_1 \qquad [4]$$

and the actual temperature rise in any case is this value divided by the compressor efficiency. This is the source of the very convenient and commonly used expression for compressor efficiency, otherwise known as the "temperature-rise ratio"

$$e_t = \frac{YT_1}{T_2 - T_1} \qquad [5]$$

which should be equal to the actual shaft efficiency if there is no heat transfer to or from the surroundings.

Equation [4] and Fig. 4 can now be used to determine the temperature rise of the air in the foregoing examples of supercharger power. For the 32-in. manifold pressure at 10,000 ft

$$T_2 - T_1 = YT_1/0.7 = 65/0.7 = 93 \text{ F}$$

and for the 40-in. pressure at 25,000 ft

$$T_2 - T_1 = 188/0.7 = 269 \text{ F}$$

With an inlet-air temperature of —30 F, the manifold temperature in the latter case would be 239 F. This would be intolerably high, according to present standards, for reasons which will be discussed later. In practice, the total pressure ratio of 3.6 involved in this case would be divided between two stages of compression with an intercooler between them, as indicated in Fig. 5. With the conditions assumed in this illustration the manifold temperature would be reduced to 134 F.

Referring further to Fig. 5, the first-stage discharge pressure is commonly set at 31.67 in., in order to maintain standard sea-level pressure, 29.92 in., at the carburetor inlet, allowing for a drop of $1^3/_4$ in. Hg in the intercooler and ducting. The temperature rise in each stage was obtained from Equation [4], allowing for a compressor efficiency of 70 per cent. It will be observed that the first-stage pressure ratio, p_2/p_1, is 31.67/11.1 = 2.85, whereas the second-stage ratio is 40/28.6 = 1.4, in order to make

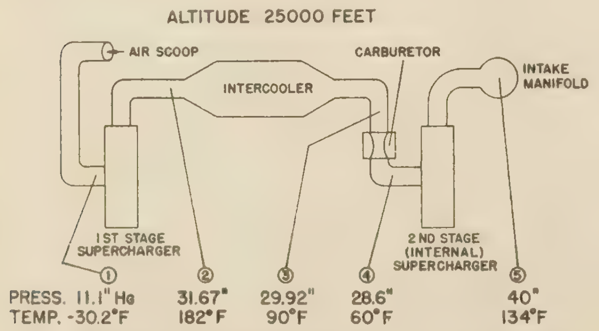

Fig. 5 Pressures and Temperatures in a Typical Two-Stage Induction System

more effective use of the intercooler. (Occasionally an aftercooler is used between the final stage and the manifold, but this is not convenient in radial engines.) The so-called "intercooler effectiveness" is the ratio of the drop in temperature of the compressed air to the initial temperature difference with respect to the ambient, or

$$\frac{t_2 - t_3}{t_2 - t_1} = \frac{182 - 90}{182 - (-30)} = 43.4 \text{ per cent}$$

which represents a compromise in performance to reduce cooler weight. The drop of 30 F in the carburetor is due to the absorption of latent heat by the evaporating gasoline. In some cases, most of this occurs in the passages of the internal supercharger, but the effect is about the same.

There are at least three reasons for limiting the manifold temperature to the lowest practicable level:

1 To avoid detonation or "knocking" in the engine, which has such a serious effect upon engine life as to constitute a major limitation to the allowable power rating. As the mixture temperature increases, fuel of higher-octane rating must be employed to maintain constant intensity of detonation.

2 To increase the density of the charge supplied to each cylinder, thereby increasing the power developed, as per Equation [1].

3 To reduce the combustion and exhaust temperatures of the engine.

These three results are promoted by improving the efficiency of the supercharger, which reduces the temperature rise during compression. In addition, any improvement in first-stage efficiency reduces the work of compression in a second stage by reducing the value of T_1 in Equation [3].

On the other hand, these same results can be promoted by increasing the effectiveness of the intercooler. In either case higher efficiency or effectiveness inevitably results in greater size and weight, at a given stage of the art. Optimum installation design therefore calls for a careful compromise of these factors to obtain best over-all performance, with due regard for total drag, power, and supercharger operating speed.

Types of Superchargers

Two basic types of compressors have been used to supercharge internal-combustion engines:

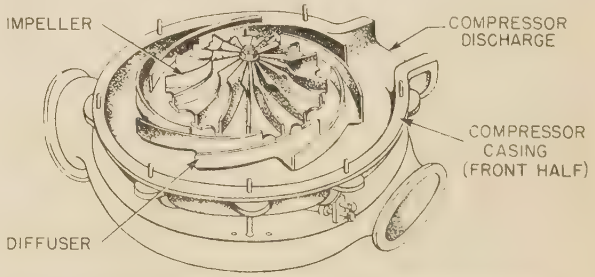

Fig. 6 Centrifugal Compressor in Turbosupercharger

1 The "positive-displacement" type, of which the Roots blower is a prominent example. This has two rotors carrying lobes which mesh with each other as do gear teeth, and which during their rotation open and close a displacement space between the dumbbell-shaped lobes. An excellent description of this blower was given by John L. Ryde in a recent S.A.E. paper (6). It has not been applied to the supercharging of airplane engines to any appreciable extent since the N.A.C.A. tests of 1930, reported by Oscar Schey (7). Other positive-displacement pumps,

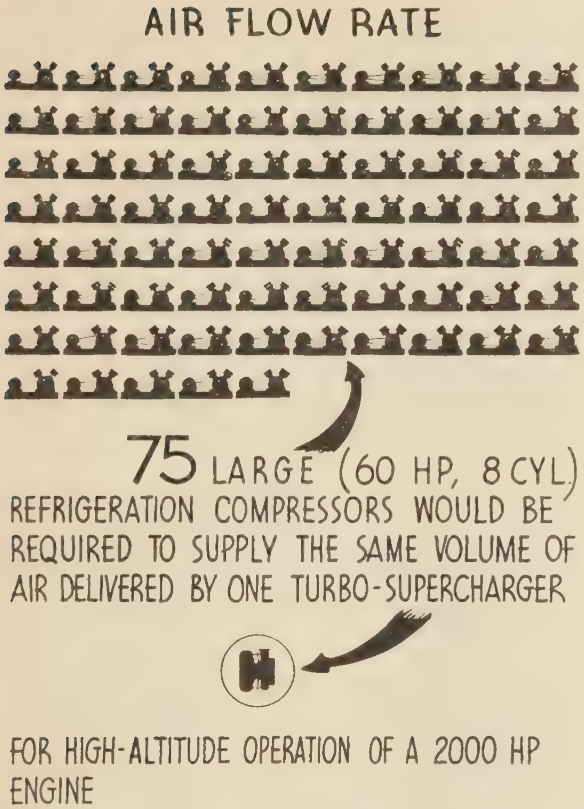

Fig. 7 Contrast Between Supercharger and Equivalent Reciprocating Compressors

such as the rotary-vane and reciprocating-piston types, have not been well adapted for use in aircraft, principally because of their relatively large size and weight for a given capacity.

2 The "centrifugal" type, the principal elements of which are a multivaned impeller and an annular diffuser, Fig. 6, enclosed in a suitable casing, which includes a collector or scroll surrounding the diffuser. The impeller is driven at speeds of the order of 20,000 to 30,000 rpm, with peripheral velocities or "tip speeds" commonly reaching 1100 or 1200 fps. The impeller imparts energy to the air principally by giving it violent peripheral and angular acceleration, roughly one half of this energy appearing as static pressure rise and half as kinetic energy at the exit (8). The function of the diffuser is to convert as much as possible of the velocity of the high-speed air streams into further increase in the static pressure, by a process of efficient deceleration. Because of its high speed of operation, the centrifugal supercharger has an exceptionally high volumetric capacity, which makes it especially well adapted for use in aircraft. By way of contrast, Fig. 7 shows graphically the relative size of a centrifugal supercharger (turbine-driven) as compared with the 75 conventional electrically driven reciprocating compressors which would be required to furnish the air for a 2000-hp engine.

The "axial-flow" compressor should be mentioned as another basic type, which has frequently been considered for supercharging engines (9). It is essentially a series of propeller-type fans, as many as 16 or 20, mounted on the same shaft, with a set of fixed turning vanes between stages, not unlike a multistage steam turbine. One of its limitations is its relatively narrow operating range, which has the character of curve B in Fig. 3, except that its peak efficiency may be appreciably higher. Its chief disadvantage is its large size, principally in axial length, due to the many stages required to produce a substantial pressure ratio. Mainly for these reasons, it has not been applied to aircraft engines, although it has been used commercially in gas-turbine power plants (10, 11).

Centrifugal-Compressor Design Factors

In selecting a centrifugal compressor for a given supercharger application, the following factors must be taken into account: (*a*) Capacity, (*b*) maximum pressure ratio, (*c*) efficiency, (*d*) range, (*e*) impeller speed, (*f*) mechanical ruggedness, (*g*) size, (*h*) weight.

It is, of course, impracticable to discuss all of these factors exhaustively in a paper of this kind, but a brief survey of some of the major effects may be helpful to the uninitiated. More detailed discussions of the basic theory and practice will be found in references (8) and (12) of the Bibliography.

Capacity, usually expressed in pounds of air per minute, is governed primarily by impeller diameter, D, and revolutions per minute, N. For a series of "similar" machines, the volumetric flow Q (cfm) is proportional to ND^3, so that all sizes of such a series should have the same value of Q/ND^3. The basic capacity of the design, or "characteristic flow," which Q/ND^3 represents, can be increased by increasing the depth of the impeller passages or the "relative" area of the inlet. Any considerable departure from optimum proportions, however, will impair the efficiency.

As previously indicated, a supercharger must deliver 0.12 lb of air per min per engine hp. The volume flow at any altitude is obtained by dividing the mass flow by the density, to obtain the value of Q. The machine must then be selected or adjusted to bring the rated value of Q/N to the appropriate point near the peak of the characteristic curve, Fig. 3. In practice, the final adjustment of the capacity of the compressor is accomplished by changing the angle and the flow area of the diffusers suitably. (This is done by changing the entire diffuser, which is nonadjustable.) The volume flow required to maintain a given capacity increases rapidly with altitude, becoming 6.6 times as great at 50,000 ft as at sea level.

The maximum pressure ratio is an important criterion of the excellence of a given design, as it fixes the useful ceiling of the machine. The pressure-producing capacity of a centrifugal compressor is measured by the so-called "pressure coefficient"

$$\eta = \frac{6083YT_1}{V^2} \qquad [6]$$

where V is the impeller tip speed in feet per second. This coefficient compares the energy of isentropic compression of the air between the actual pressures involved in Y, with the energy theoretically imparted to the air by the impeller (which is proportional to V^2). From Equation [6], the necessary operating speed to deliver the required pressure ratio can be obtained as

$$V = \sqrt{\frac{6083YT_1}{\eta}} \qquad [7]$$

the value of YT_1 being taken from Fig. 4.

In general, pressure coefficients for impellers with straight radial blades vary with Q/N in about the same manner as the efficiency (e_i), as shown by the curves in Fig. 3. Although Equations [6] and [7] do not apply rigorously to impellers having curved blades, it may nevertheless be said that forwardly curved blades produce relatively high pressure coefficients, in that a lower tip speed is required for the same pressure ratio. This is secured, however, at the expense of a considerable reduction in stability and range of operation. Conversely, impellers with backwardly curved blades commonly show higher efficiencies but

lower pressure ratios for the same tip speed than the straight-bladed type.

Altitude performance depends upon the ability to develop pressure ratio with a minimum temperature rise over a wide range of stable operation, and therefore different types of superchargers should be compared on this basis, rather than at the same tip speed, as has sometimes been the practice in the past.

Substituting the value of Y in Equation [6], it can be shown that

$$p_2/p_1 = \left(1 + \frac{\eta V^2}{6083T_1}\right)^{3.534} \qquad [8]$$

The maximum pressure ratio which a given machine can develop is usually limited by the allowable stresses due to centrifugal force, which is proportional to the square of the speed. But, if this is not the limiting factor, there is a general tendency for the pressure coefficient and range to decrease at high speeds, as indicated by the change from curve A to curve B in Fig. 3. The ultimate ceiling may be reached at the point where the value of η collapses so that there is no further increase in pressure with speed.

The efficiency of a centrifugal compressor depends almost entirely upon how successfully the air is introduced into the impeller and conducted through the passages of the machine without shock, flow separation, or turbulence. This sounds much simpler than it actually is. The air entering the impeller is given a sudden terrific acceleration, sometimes reaching a figure of 6,000,000 fpsps, or 186,000 g. The internal air velocities are commonly of the order of 1000 to 1200 fps at the diffuser entrance. The latter figure would exceed the velocity of sound (about 1088 fps at 32 F) except for the fact that this velocity increases in proportion with the square root of the absolute temperature, which increases very considerably with the speed. At these speeds, the air particles tend to travel through the flow passages like rifle bullets, so that it is very difficult to make them turn corners and fill the varying flow sections without separation.

At the higher air velocities associated with relatively high-volume flow and tip speeds the performance of the compressor is usually limited by the occurrence of "shock waves" manifested by very sudden changes in pressure with high energy losses, as the velocity at the diffuser entrance approaches the sonic or acoustic value. The tendency of these shock losses to occur is measured by the so-called "Mach number," which is simply the ratio of the air velocity, u, to the velocity of sound, C, at the local temperature (or any convenient reference temperature). Since the critical value of u is determined by and is very nearly equal to the tip speed, V, and since $C = 49\sqrt{T}$ it is convenient to express the local Mach number as

$$M = \frac{V}{49\sqrt{T}} \qquad [9]$$

Unfortunately, the local value of T at the diffuser inlet is usually not known, but for a given value of V it is determined by the inlet-air temperature, T_1, so that it is still more convenient to use a distorted Mach number

$$M_1 = \frac{V}{49\sqrt{T_1}} \qquad [10]$$

in comparing the basic efficiencies of different compressors. This means that all of the values of efficiency versus Q/N should fall on the same curve, as A in Fig. 3, for a constant value of M_1 no matter how the speed and inlet temperature may be varied. A higher value of M_1 would tend to reduce the performance in the direction of curve B.

Since efficiency, pressure coefficient, and range are thus impaired by the high speeds and low temperatures required for high-altitude operation it is customary to employ two stages of compression when the total pressure ratio is greater than about 2 or 3 to 1. However, there is always a strong incentive to develop single-stage designs to yield the utmost in pressure ratio without too great sacrifices in performance, for reasons of mechanical simplicity, weight, and size.

The significance of the problem of operating range has already been indicated. The problem is becoming progressively more acute as the spread between maximum and minimum engine power increases, and as higher altitudes call for higher supercharger speeds. One expedient for increasing range is the use of a ring-type or vaneless diffuser, which gives the type of performance represented by curve C of Fig. 3. However, the penalty is usually a substantial increase in size (12), or else a loss in efficiency and pressure coefficient (13).

Speed, mechanical ruggedness, and weight are closely interrelated factors. The high speeds at which centrifugal superchargers operate not only set up high centrifugal stresses in the impeller but also generate vigorous vibrations in all parts of the machine, due to resonance effects and the impulses caused by the passage of impeller blades past the diffuser vanes. The energy and destructive power of these vibrations is sometimes surprising, and it requires very special care and experience to determine the necessary massiveness and structure to prevent premature fatigue failures.

For aerodynamic reasons, to preserve the proper proportions of the flow passages, the capacity of a given type of supercharger should not be increased faster than the square of the characteristic dimension, which is usually the diameter, i.e.

$$Q \propto D^2$$

In any series of similar machines, the weight increases with the cube of the characteristic dimension, or

$$W \propto D^3$$

Hence

$$W \propto Q^{3/2}$$

which indicates that the larger-capacity machines tend to become relatively heavy. This aspect of the problem becomes so serious that there is a strong incentive to cramp and compromise the design in order to maintain more nearly proportionate weights as the horsepower rating is increased. For this reason, the impression is sometimes given that the art is not progressing, since very real gains in basic merit are required to stand still with respect to efficiency while constantly progressing in capacity and altitude rating.

Types of Drives

The commonest type of supercharger, embodied in practically every major commercial and military aircraft engine today, is the radial centrifugal gear-driven from the rear end of the engine crankshaft. The speed is usually stepped up through two sets of gears to give impeller speeds of from 6 to 10 times the crankshaft speed.

For conservative performance at moderate altitudes, as in typical air-line service, a single-stage single-speed drive is adequate, but for high-performance military planes, it has been found expedient to employ some form of multispeed drive, for the following reasons:

If a centrifugal supercharger is operated at constant speed the pressure ratio and volumetric capacity are substantially independent of the initial pressure and density of the air. But the power is proportional to the weight flow, and hence to the density, and of course the final pressure increases in proportion with

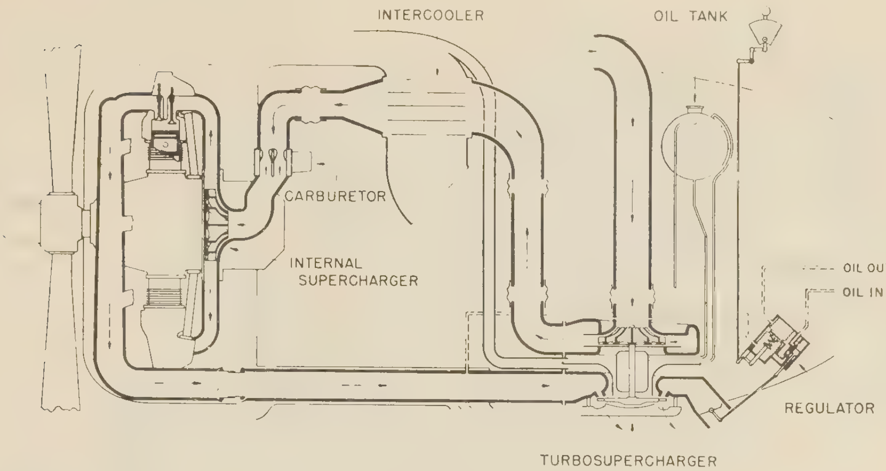

Fig. 8 Schematic Diagram of Turbosupercharged Power Plant

the initial pressure. A supercharger designed to give a full-power manifold pressure of 45 in. Hg at 25,000 ft, where the pressure is 11.1 in. and the density is 0.0343 lb per cu ft, would therefore give a manifold pressure of 121 in. at sea level, and the power required to drive it would be 2.23 times normal. To avoid wrecking the engine immediately, the throttle would have to be closed to drop the manifold air pressure from 121 in. to 45 in. But the supercharger would still extract an unnecessarily large block of power from the engine, and the heat of compression, starting with a relatively high initial air temperature, would tend to cause heavy detonation unless the manifold air pressure and engine power were reduced still further. In order to maintain the maximum net engine power or economy at low altitudes it is therefore necessary to regulate the supercharger speed so that no more than the required pressure is generated at each level. In fact, it is one of the axioms of good flight engineering that the throttle should be kept open as far as practicable under all operating conditions.

Gear-driven superchargers, either single or two-stage, are now frequently provided with a two-speed drive having a mechanical clutch for shifting from low- to high-gear ratio. More recently, fluid couplings have been introduced, particularly in German engines, to provide continuously variable-speed control. An excellent description of the details of these drives is contained in a recent article by F. M. Kincaid of the Wright Aeronautical Corporation (14). All of them are characterized by a relatively high degree of mechanical complexity, especially when fully automatic control is provided to maintain the desired manifold air pressure independently of altitude. The variable-speed feature of the fluid couplings is achieved at the expense of considerable waste power, which must be dissipated in the form of heat in the oil coolers.

The exhaust-gas-driven turbosupercharger represents a very practical solution to the drive problem in terms of mechanical simplicity, flexibility, and power economy. As may be seen from the schematic diagram, Fig. 8, the impeller of the turbosupercharger is direct-connected to a single-stage turbine driven by the engine exhaust gas. The speed is controlled in a very simple manner by allowing excess gas, not required for turbine operation, to escape through the wastegate instead of through the turbine nozzles and wheel. This makes it possible to hold constant manifold pressure, with no throttling and no extraction of engine crankshaft power, from sea level to critical altitude. No more than the minimum required power is drawn from the turbine at any time, so that the excess back pressure which it imposes upon the engine is likewise never more than sufficient to supply the desired manifold air pressure. The significant fact here is that the reduction in engine power caused by this increased back-pressure is usually much less than the crankshaft power required to drive an equivalent geared supercharger (15).

As indicated in Fig. 8, it is common practice to employ an internal-geared supercharger in conjunction with the turbosupercharger, with an intermediate intercooler. With a relatively low pressure ratio, as in the system, Fig. 5, the internal stage has no speed control, since it will not generate excessive manifold pressures with open throttle at sea level if the turbo is cut out.

The General Electric Turbosupercharger

Figs. 9 to 12 show various views of the General Electric turbosupercharger of the type used in the Boeing Flying Fortress, Consolidated Liberator bomber, and the Lockheed Lightning and Republic Thunderbolt pursuits. Although military necessity prevents the disclosure of complete technical details, the major features may be indicated as follows:

Referring to the cutaway view, Fig. 9, the engine-exhaust stack connects to the nozzlebox *A*, which is directly above the turbine wheel *B*. The hot exhaust gases expand through the nozzles *C* and impinge upon the turbine buckets *D*. The pressure in the nozzlebox is regulated by means of the waste gage *E*, which vents excess gas to the atmosphere.

The power developed is transmitted through the shaft *F* to the impeller *G* (which is also shown in Fig. 6). The diffuser passages *H* discharge into a scroll-type collector in the outer portion of the compressor casing *J*, which has a radial outlet. The rotor assembly is carried in a ball bearing *K* at the impeller end to take the thrust load, and a roller bearing *L* at the turbine end to allow for expansion of the shaft.

The baffle ring *M* serves to shield the compressor casing from the heat of the nozzlebox, and its inner portion supports the nozzle ring.

The lubricating-oil pump *N* is driven by a worm gear from a worm sleeve keyed to the shaft. The pump is of the gear type, and actually consists of two separate pumps within a single casing. One element supplies oil under pressure to the gears and bearings. The other element is a scavenging pump which removes oil from the lower part of the bearing housing *P* and returns it to the supply tank.

The external appearance of the turbosupercharger is shown

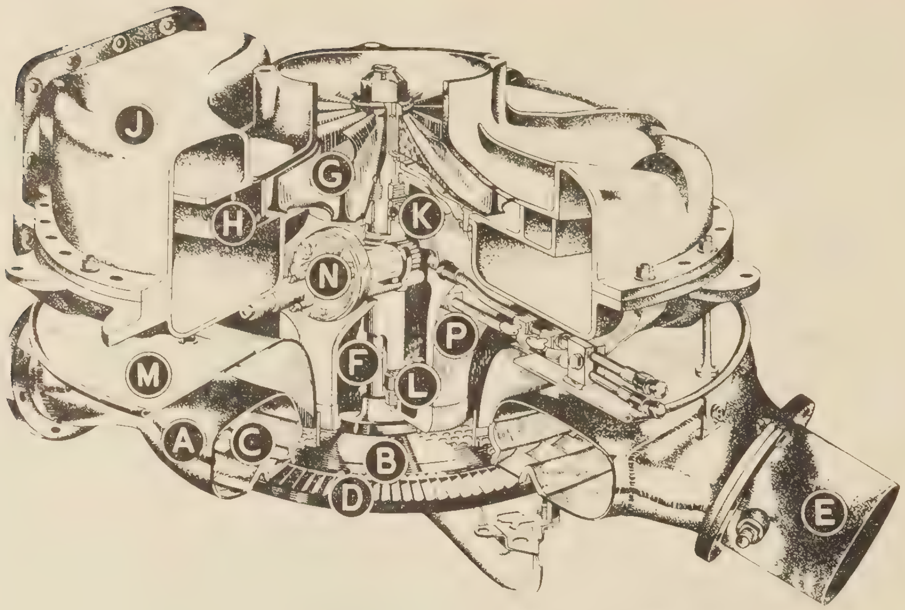

FIG. 9 CUTAWAY VIEW OF TURBOSUPERCHARGER

in Figs. 10, 11, and 12. In Fig. 10, the cooling cap has been removed to expose the turbine wheel.

The impeller is machined out of an aluminum-alloy forging. The compressor casing, diffuser, and bearing are aluminum-alloy castings. Turbine parts are made of alloy steels especially developed to withstand the high stresses, elevated temperatures, and corrosive action of the exhaust gases. Data on various materials of this latter class are given in reference (16).

As indicated in Fig. 8, regulation of the turbine waste gate is effected by means of a hydraulic servomotor containing a pressure-sensitive element connected to the engine exhaust stack ahead of the turbine. By adjusting the boost-control lever, the pilot sets the regulator to hold a constant nozzlebox pressure which will maintain the desired engine-intake-manifold pressure.

THERMODYNAMIC FEATURES OF TURBODRIVE

The development of the gas-turbine drive is described in references (16, 17, 18), and in a companion paper by Dr. Sanford A. Moss (19). The underlying thermodynamic principles are essentially the same as those applying to the conventional single-stage impulse-type steam turbine. The success of the turbosupercharger is based upon the development of an adequate mechanical design to enable the turbine to utilize the energy of the high-

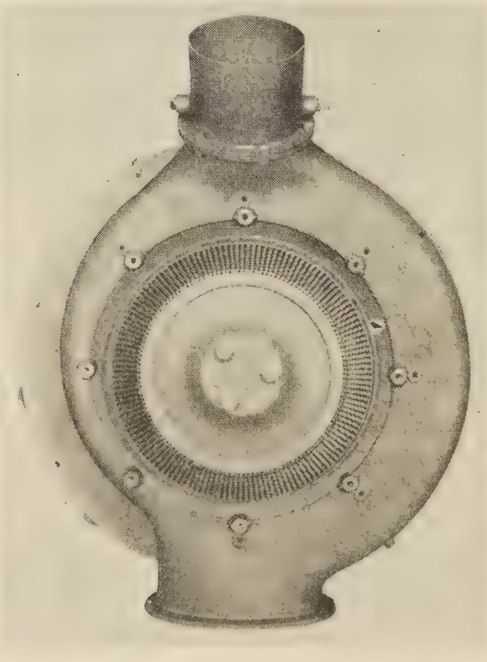

FIG. 10 WHEEL-END VIEW OF TURBOSUPERCHARGER

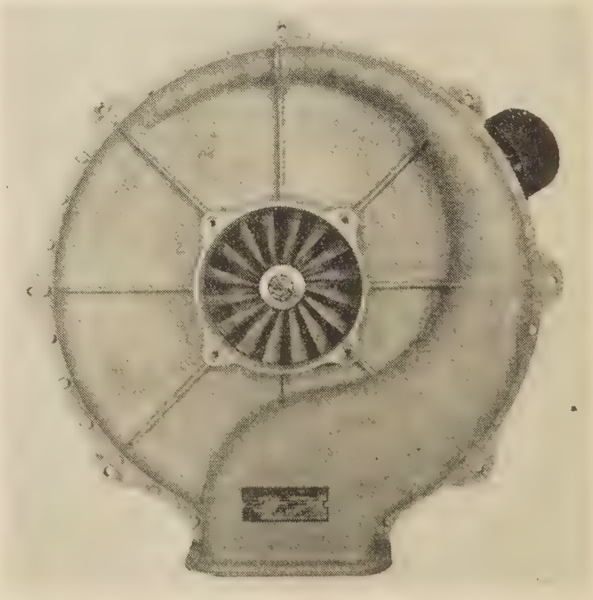

FIG. 11 COMPRESSOR-END VIEW OF TURBOSUPERCHARGER

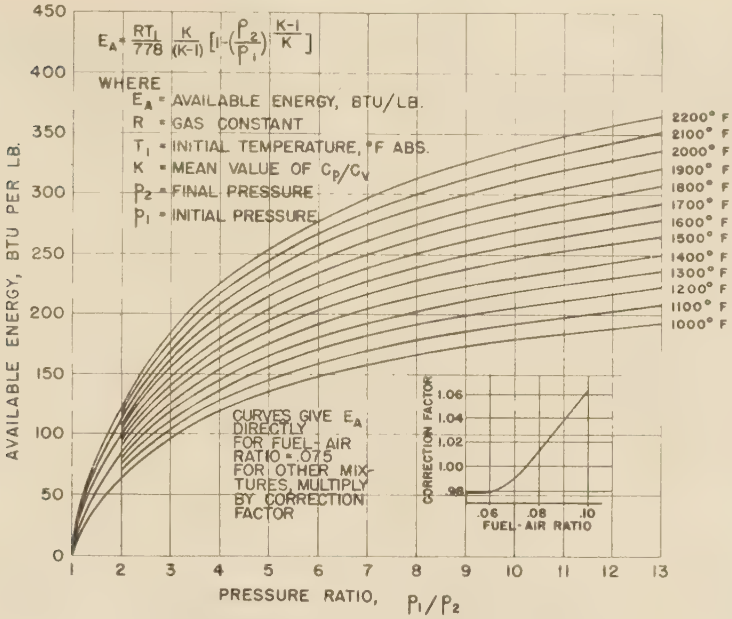

Fig. 13 Available Energy of Hot Exhaust Gas Versus Temperature, Pressure Ratio, and Fuel-Air Ratio
(Figures on curves are ordinary degrees F.)

temperature engine exhaust gases with satisfactory reliability and life and without undue penalties in weight or size.

Fig. 13 gives the available energy, in Btu per pound, of engine-exhaust gas for various temperatures, pressure ratios, and fuel-air ratios. The data are based upon recent information on the specific heat of gases reported by Heck (20), and Ellenwood, Kulik, and Gay (21), and are believed to be considerably more accurate for this purpose than the energy values based upon the properties of ordinary air. (To convert the available energy to hp per lb of gas per min, divide Btu per lb by 42.4.) It should be noted that this is primarily thermal energy, representing the waste heat of the exhaust gases and that the pressure ratio required to release a given amount of power decreases rapidly as the initial gas temperature increases.

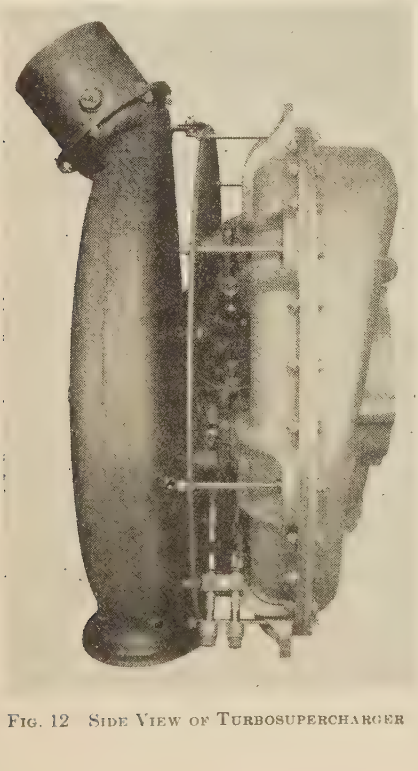
Fig. 12 Side View of Turbosupercharger

Actual exhaust-gas temperatures, in the case of high-powered military engines, vary from about 1200 to 1700 F, the higher figures being associated with higher outputs and leaner fuel-air ratios. Assuming a representative temperature of 1500 F, it is of interest to determine the available energy per pound of gas for a pressure ratio of 2.69, which corresponds to sea-level back pressure (p_1 = 29.92 in.) in the engine-exhaust stack at 25,000 ft (p_2 = 11.1 in.). The conservative "cruising lean" value of 0.075 will be used for the fuel-air ratio, although richer mixtures would be used for full power.

From Fig. 13, for p_1/p_2 = 2.69 and T_1 = 1500 F, the available energy is 122 Btu per lb, or 2.88 hp per lb per min. This figure must be multiplied by the turbine efficiency to obtain the actual shaft output.

The engine requires slightly less than 1 lb of air per lb of exhaust gas. It will be assumed that the turbocompressor delivers this air at sea-level pressure at the carburetor under the conditions of Fig. 5 (p_2 = 31.67) with a compressor efficiency of 70 per cent. From Equation [3] and Fig. 4, the required power input to the compressor will be

$$0.00573 \times 148/0.7 = 1.21 \text{ hp per lb per min}$$

This means that the turbosupercharger can maintain sea-level pressures at both ends of the engine at 25,000 ft so long as the turbine efficiency is greater than 1.21/2.88 = 42 per cent. This is entirely practicable, even after allowing for losses due to gas leakage, etc., and is the basis of most turbosupercharger applications.

As the altitude increases, the compressor must develop a greater pressure ratio, which requires more power. But with constant engine inlet and exhaust pressures, the pressure ratio across the turbine also increases with altitude, so that the power available

continues to balance the power required. The ultimate level above which this balance can no longer be maintained is determined either by the attainment of the maximum allowable "turbo" speed, or else by loss of efficiency at excessively high pressure ratios. Beyond this critical altitude, the carburetor pressure and engine power begin to decline in the normal manner.

From the energy standpoint, it might be said that the turbo thrives on high exhaust-gas temperatures, as indicated by the curves in Fig. 13. However, at temperatures in the vicinity of 1700 F the cooling problem becomes very serious; to such an extent that it has been thought in many quarters, particularly abroad, that turbine operation at such temperatures was impracticable. The fact is that the ability of the turbine materials to withstand these high temperatures is due largely to the fact that no parts of the machine actually reach the full temperature of the gas. Practically all parts are cooled by intense radiation to the surroundings, and the nozzlebox is cooled about 400 deg below the gas temperature by the additional effect of a stream of "rammed" air. The effective temperature of the gas impinging upon the buckets is reduced several hundred degrees by virtue of the conversion of thermal energy into kinetic energy in the high-velocity jets (see reference 22). It appears, therefore, that present engine-exhaust temperatures are very nearly at the optimum level for turbo operation; if they were much higher the necessary reduction in turbine speed or life would be serious, and if they were much lower the performance would be impaired by the reduction in available power.

The problem of "afterburning" may be mentioned in this connection. Since aircraft engines are always operated with rich fuel-air mixtures at high powers, the exhaust gases contain considerable quantities of carbon monoxide and hydrogen; up to 14 per cent CO and 7 per cent H_2 (23). These highly inflammable gases will burst into flame if they come into contact with air before cooling below their ignition temperature (roughly 1100 to 1200 F). Because of their visibility at night, these flames must be suppressed in the case of bombers, many of which are provided with bulky quenching devices for this purpose. In turbosupercharger installations afterburning can cause local overheating if the gases come in contact with sufficient air, by entrainment or otherwise, before passing completely through and away from the turbine. But with suitable precautions to avoid this, the turbosupercharger has a salutary effect in helping to suppress visible flames by cooling the gas several hundred degrees, due to the conversion of heat into work.

Comparative Performance of Superchargers

The first criterion of supercharger performance is the ability to maintain full engine brake horsepower at all altitudes in the operating range of the airplane. The final measure is, of course, the actual performance of the plane, including rate of climb and maneuverability as well as speed in level flight.

The effect upon engine power can be determined or predicted a great deal more readily and accurately. Fig. 14 gives the results of a very interesting study of the performance of an engine equipped with various types of superchargers, recently published by the Allison Division of General Motors Corporation (24). Curve 1 represents a "sea-level engine" which has a small low-speed internal supercharger provided to produce full power by ground boosting the manifold pressure at sea level with full throttle. With a fixed impeller speed, the engine power is the same as that of the unsupercharged engine shown in Fig. 1. Curve 2 shows the result of speeding up the internal supercharger so as to produce maximum power at a rated critical altitude of 15,000 ft. This engine must be partly throttled at sea level to avoid excessive manifold pressure and detonation. The net output is reduced by the extra compressor power and temperature rise due to the

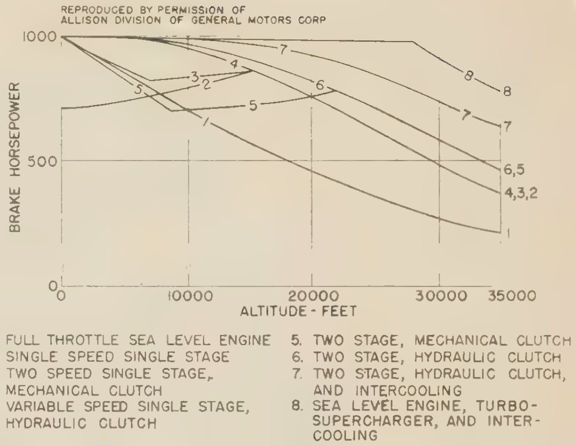

Fig. 14 Altitude Performance of an Engine With Various Types of Superchargers

higher impeller speed, as discussed in the section "Types of Drives." As the plane rises, the engine power increases, due to the reduced back pressure plus the favorable effects of the lower inlet-air temperatures, until the throttle is fully open at 15,000 ft.

The remaining curves are determined by similar considerations of the shaft power required to drive the compressor, and the effects of manifold air temperature upon detonation and charge density. As just mentioned, the outstanding performance of the turbosupercharger (curve 8) is due to its ability to supply exactly the required pressure with no extraction of crankshaft power. The principal adverse effect of the turbosupercharger is to prevent the engine back pressure from following the normal decline of ambient pressure at altitude. Although curve 8 shows a slight loss of power with altitude, it has been demonstrated by flight tests that it is entirely practicable and convenient to hold 100 per cent sea-level power or more all the way up to the critical altitude with a conventional turbosupercharger installation.

On the other hand, the gear-driven supercharger, particularly the simpler single-speed types, are considerably lighter and more compact. If there is no intercooler, the drag will be less. If a turbosupercharger is not used it is possible to convert more of the exhaust energy into thrust by the use of rearwardly directed ejector-type exhaust manifolds (25), although the efficiency of this process is low.

It is extremely difficult to make valid comparisons of the relative performance of a given airplane with a gear-driven supercharger or with a turbosupercharger for the simple reason that, for optimum results, the plane must be designed specifically for the particular installation used. This is especially true in pursuit ships, where the supercharger has a marked effect upon fuselage shape and weight distribution. Moreover, the requirements of a light "hedge-hopping" plane are apt to be quite different from those of a heavy, high-altitude design. The performance of any supercharger, and particularly the turbo, depends so greatly upon the excellence of the installation that it is entirely possible to reverse the results of any such comparison by suitably butchering the job. It is significant, however, that the gear-driven superchargers have had the benefit of much more intensive work on the installation problem, so that some account should be taken of the potential as well as the actual merit of the turbo type. Even the latest and best turbo installations are not as closely and completely integrated into the power plant as they might be. But the fact remains that the turbosupercharger enjoys an initial and basic thermodynamic advantage, as indicated in Fig. 14, which becomes even more marked at higher altitudes. Finally,

it is still more significant that four of the outstanding planes that this country has contributed in the present war, the B-17, B-24, P-38, and P-47 are turboequipped.

Present Status of Turbosupercharger

The question is sometimes asked: Why is it that no foreign country has used the turbosupercharger in military airplanes during the present war? The answer is that, although a considerable amount of experimental work has been carried on abroad as described by von der Nüll (16), the practical difficulties of design and production have discouraged any large-scale application to high-performance Otto-cycle engines. The Germans have succeeded in applying a turbo to their Junkers Jumo 207 Diesel on a small scale, but since the exhaust-gas temperature entering the turbine is about 1000 F, the performance is correspondingly low. The outstanding success of the turbo development in this country has resulted from the sustained effort and continuous experience with the manufacture and application of this type of supercharger since 1917 which has been made possible by the support and cooperation of the United States Army Air Forces. Consequently, at the outbreak of the war this country alone had both the technical "know-how" and the facilities for producing these machines in the necessary quantity and quality. It was in recognition of this achievement that the Collier trophy was awarded jointly to the Army Air Corps and Dr. Sanford A. Moss, of the General Electric Company, by the National Aeronautic Association in 1941.

For several years the turbosupercharger has been in mass production, at a rapidly increasing rate. At the present time it is being produced at the rate of thousands a month in five large plants, two of them especially built for this purpose. The Ford Motor Company and the Allis-Chalmers Manufacturing Company are contributing in a substantial degree to the production of the General Electric design.

Since the start of the war large numbers of turboequipped planes have seen active service in the battle areas all over the world. All have given an excellent account of themselves. The turbo itself has achieved a remarkable record of reliability and outstanding performance under an extreme range of operating conditions. Many favorable reports have also been received in respect to the relative ease with which it has been serviced and maintained in the field. The most enthusiastic tributes, however, have come from pilots and crews who have repeatedly been able to outclimb and outdistance enemy planes whose engines weaken and falter in higher altitudes.

Future Prospects of Supercharging

As in many other branches of aviation, the war has given a tremendous impetus to the development of all forms and applications of supercharging. All known areas of theory and practice are being explored searchingly and extensively and new fields are being opened up under the incentive to improve the performance and extend the capacity of existing machines. Unfortunately, it is obviously impossible to discuss the results that are being accomplished. About all that can be said is that the general direction is up, and that "the sky's the limit." It might be added that superchargers are definitely ahead of engines, planes, and propellers in this direction, at the present time.

With reference to postwar supercharger applications, it is confidently anticipated that turbos will be applied to commercial transport planes, particularly for the longer, high-altitude runs with definite advantage in operating economy. Although there is a lack of adequate flight data to support this point there are a number of sound reasons, some of which have been cited herein, upon which to base the expectation. Von der Nüll (16) mentions some prewar German tests with "modern high-altitude engines," showing specific fuel consumptions of 0.441 lb per hphr with a turbosupercharger, as compared with 0.584 lb per hphr with a gear-driven supercharger, but the details are not given.

The advantages of high-altitude flight (3) which led to the development of the famous Boeing Stratoliners will doubtless encourage the development of pressure-cabin planes for operation in the higher levels of the stratosphere. Very much more adequate cabin-supercharging systems will then be available. Here again the basic economy and flexibility of the turbocompressor can be utilized effectively. Since it delivers pure air at about the desired pressure and temperature, the engine turboinstallation can be used to pressurize and ventilate the cabin by bleeding off a small fraction of the air leaving the intercooler. The internal-type supercharger cannot be so employed because it discharges a mixture of fuel and air.

But whatever the future may bring, it can be stated with assurance that the supercharger in its various forms will be ready and adequate for the demands of new advances in high-altitude flight, for without it such flight as we know it today would have been impossible.

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Discussion

Kenneth Campbell.[3] The paper constitutes a technical exposition of the basic principles, characteristics, and functioning of superchargers as applied to aircraft engines. It is clearly and comprehensively expressed, and the authors are to be congratulated on setting forth this condensed explanation of the subject as a whole. It is needed by the thousands of engineers in aeronautics whose assignments are nearly always concerned in some respect at least with the subject of supercharging and its many ramifications.

Without doubt, the authors would be the first to agree that the material of this paper is necessarily largely not new, measured by the very short time standards of aeronautical development, since everything in the field of aeronautics which is new, judged by these standards, is of use to the enemy and must therefore be left unsaid. The writer is privileged to be familiar, however, with the large amount of manpower and expenditure being devoted to physical research in the field of supercharging by the authors' organization, as well as with some of the results being obtained, and the reader can rest assured that the information given in the paper rightly comes from the oldest and one of the most active centers of supercharger development.

The paper is fundamentally sound, and we have little worthy comment to make taking issue with it technically. The authors recognize the existence of some exhaust energy from rearwardly ejected exhaust manifolds. It might be pointed out, however, that the thrust horsepower gain to the airplane from simple, rearwardly pointed exhaust stacks from each cylinder has for some years been calculated by many investigators to equal the mechanical power robbed from the crankshaft by the mechanically driven first stage, even in climbing flight. On the other hand, gain from directing the turboexhaust to the rear is necessarily slight. If these calculations be true, the comparison of curve 8 in Fig. 14 of the paper, representing the turbo performance, with the other performance curves of this figure will be very misleading to the uninitiated.

We enjoy and approve highly the statement made in the paper that the impression is sometimes given that the art of supercharging is not progressing, since very real gains in basic merit are required to stand still with respect to efficiency, while progress is constantly being made in altitude and capacity rating. During the period of supercharger knowledge covered by this paper, superchargers have increased from 500-engine-horsepower capacity to 5 or 6 times that capacity. As the authors indicate, in order to have maintained the efficiencies of the small superchargers in the larger models without further performance research, impeller diameters and other corresponding dimensions should have more than doubled; yet no such maintenance of dynamic similarity has had to take place. The fact that the efficiencies of superchargers of engines of 2000 hp are slightly better, and in some cases substantially better, today than the efficiencies obtained originally with superchargers for engines of 500 hp is the result of very extensive research work which has been going on in various localities in this country in supercharger development, without which the superchargers of large engines today would be failures. This is a point seldom appreciated in the past and one which the authors do well to express so clearly.

[3] Wright Aeronautical Corporation, Paterson, N. J.

J. M. Heldack.[4] It is extremely fortunate that the military authorities have allowed the presentation of such a timely subject, especially now when the nation's newspapers are full of the successes of our military airplanes, due especially to their superior high-altitude performance.

The authors have mentioned the problem of covering adequately the increased operating range of modern military-aircraft engines. Military operation requires both high-power and cruising-power operation at all altitudes up to and including the critical altitude of the installation. At present, the turbosupercharger is the only means of supercharging which allows this flexibility. It can be reasonably expected that postwar application on long-range transoceanic and transcontinental passenger and cargo airplanes will be predicated on turbosuperchargers designed to operate at their maximum efficiency while covering the cruising-power range of the engine. Such a combination allows the attainment of optimum conditions from the standpoint both of performance and weight saving.

It is not generally realized that the most economical engine revolutions (rpm) do not necessarily result in the most economical cruising speed for the airplane. The airfoil characteristics of the plane must also be considered. Normally, under given conditions of altitude and weight, there is a definite optimum point where increasing power slightly, even though it increases fuel consumption, results in a greater increase in aerodynamic efficiency to the extent that the over-all result is an increase in the ratio of miles per gallon.

Basically it becomes a problem of matching the plane speed, endurance, and economy to result in optimum performance. The flexibility of the turbo, and its resultant beneficial effect on economy, makes it an important factor to be considered in designing the aircraft power plant for postwar commercial airplanes.

Authors' Closure

The authors greatly appreciate Mr. Kenneth Campbell's generous comments, coming as they do from such a well-qualified source.

Mr. Campbell raises a controversial question of long standing as regards the extent to which the extra thrust from individual exhaust jets can be made to compensate for the extra power required from the crankshaft to drive the internal supercharger when a turbo is not used. The issue has remained controversial for so long chiefly because the actual result depends upon the conditions assumed in the comparison. The exhaust jets show up to best advantage in the case of high-speed pursuit planes at moderate altitudes, especially when fuel economy is of secondary importance, whereas the turbo is better adapted to heavy long-range bombers and transports or to any type of plane in the altitude levels above 30,000 ft. Even in the fighter class, the high-altitude performance of the P-38 and P-47 demonstrates

[4] General Electric Company, San Diego, Calif. Jun. A.S.M.E.

the fact that the initial advantage of the turbo in respect to actual brake horsepower, as shown in Fig. 14, can be realized in practice, quite apart from calculations. It is freely conceded, however, that the excellent results lately obtained with two-stage gear-driven installations in pursuits will help to avoid any complacency in the turbo camp.

Mr. Heldack's comments are very pertinent and are especially welcomed in view of his close contact with the practical problems of aircraft design and performance. In reference to his remarks on the most economical engine speed, perhaps the matter might be stated this way: For a given airplane, total weight, and altitude there is an optimum "thrust horsepower," determined by the aerodynamics of the plane, which will give the best ratio of airplane speed to power. For a given thrust horsepower there is an optimum engine speed, determined by the characteristics of the engine and propeller, that will give the minimum fuel consumption. Optimum over-all economy, as measured by miles per gallon, can be obtained only by the proper selection of both power and engine revolutions (rpm).

The point is that the turbosupercharger makes it possible to realize this objective over a wide range of operating conditions, since the compressor speed, and hence the intake-manifold pressure, can be controlled independently of the engine speed without the loss due to throttling.

Test and Predicted Oil-Cooler Performance

By A. L. LONDON[1] and J. I. BREWSTER[2]

Air-side heat-transfer and pressure-drop laboratory test data are presented for a shell-and-tube-type oil cooler, employing steam as the shell-heating medium (1).[3] These data are compared with the information existing in the literature. The chief discrepancy observed is the surprisingly high Reynolds number obtained for transition between laminar and turbulent flow; the dip region being in the Reynolds number range 10,000 to 4000 instead of the usually expected range of 4000 to 2000. The test results, together with available oil-side conductance data, are employed to predict oil-cooling performance of the exchanger as a function of air-flow pressure drop.

Nomenclature

THE following nomenclature is used in the paper:

A_c = tube air-flow cross section for core
c, c_a = unit heat capacity of air stream, Btu/(lb °F)
c_0 = unit heat capacity of oil stream, Btu/(lb °F)
D = tube inside diameter (air side), $D = 4r_H$, ft
e = base of natural logarithms
f = over-all friction factor, including entrance and exit losses (dimensionless)
f' = tube skin-friction factor (dimensionless)
G = tube air-mass velocity, lb/(sec ft²), lb/(hr ft²)
g = 32.2 lb mass/slug mass (proportionality factor in Newton's second law, when employing both pound-force and pound-mass units)
h = unit conductance for thermal convection, air side, Btu/(hr ft² °F)
k = unit thermal conductivity of air, Btu/(hr ft² °F/ft)
K_f = constant relating β_f and β_q (Equation [1]) (dimensionless)
K_c = contraction-loss friction coefficient (dimensionless)
L = effective tube length, air side, ft
Δp = air-pressure drop, lb per sq ft, in. water
r_H = tube hydraulic radius, air side, equals flow cross section/wetted perimeter, $r_H = D/4$ (right circular cylinder), ft
t = temperature, F
t_{a1}, t_{a2} = air temperatures at tube entrance and exit, respectively, F
t_s = saturation temperature of core heating steam, °F
t_{01}, t_{02} = oil temperature at core entrance and exit, respectively, °F
T_{a1} = absolute temperature of entering air, $T_{a1} = t_{a1} + 460$, °R
Δt_1 = temperature difference, $(t_s - t_{a1})$, or average oil temperature to entering air, °F
U = over-all unit thermal conductance oil to air, Btu/(hr ft² °F)
V = air velocity, fps
w = oil flow rate, lb per hr
β = dimensionless moduli (see following group)
Δ = denotes pressure or temperature difference
ϵ = heat-transfer effectiveness (see following group)
σ = ratio of air-flow cross section to core cross section (dimensionless)
μ = air viscosity, lb/(hr ft), (centipoises)
ρ = air density, $\rho_0 = 0.075$ lb per cu ft, standard air density

Dimensionless Moduli:

$\beta_q = \dfrac{h}{Gc}\cdot\dfrac{L}{r_H}$ air-side number of transfer units
$\beta_a = \dfrac{U}{Gc}\cdot\dfrac{L}{r_H}$ over-all number of transfer units
$\beta_0 = \dfrac{\epsilon_a G c_a A_c}{w c_0}$ heat-transfer modulus employed in oil-cooling effectiveness expression, Equation [7]
$\beta_f = \dfrac{f}{2}\cdot\dfrac{L}{r_H}$ friction modulus analogous to β_q
ϵ_a = "air-heating effectiveness," $(1 - e^{-\beta_q})$, $(1 - e^{-\beta_a})$
ϵ_0 = "oil-cooling effectiveness," Equation [7]
f, f' = air-flow friction factors (see preceding group)
$\dfrac{L}{r_H}$ = modulus expressing air-side tube geometry
$N_{\mathrm{Nu}} = hD/k$, Nusselt number, air side
$N_{\mathrm{Pr}} = \mu c/k$, Prandtl number, air side
$N_{\mathrm{Re}} = DG/\mu$, Reynolds number, air side

Description of Test Apparatus

The physical characteristics of the core tested are given in Table 1.

TABLE 1 PHYSICAL CHARACTERISTICS OF CORE

Tube material	Copper
Number of tubes	637
Tube pitch, in.	0.325
Tube layout, deg	60
Tube diameter, air side, ID, in.	0.256
Tube diameter, shell side, OD, in.	0.268
Effective heat-transfer length, air side, in.	9.0
Effective heat-transfer length, shell side, in.	8.0
Heat-transfer area, air side, sq ft	32.0
Frontal area (8.9 in. diam), sq in.	62.2
Tube-flow area, A_c, sq in.	32.8
Shell-side volume, cu in.	190
Number of shell-side passes along tubes	7
Length/hydraulic radius, air side, L/r_H	140
Tube-flow area/frontal area, σ	0.53
Tube-flow area/duct area	0.52

The core unit was installed in a 9-in-diam test duct connected to the suction side of a centrifugal blower. Air-flow control was provided by a variation of the blower speed afforded by the direct-current-motor drive. Details of the test duct-blower arrangement, together with the air-flow and pressure-drop-measuring instruments, are indicated in Fig. 1.

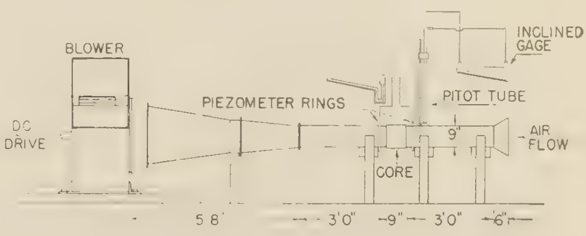

Fig. 1 Test Duct and Core Arrangement

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[2] C. F. Braun Company, Alhambra, Calif. Jun. A.S.M.E.
[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Heat Transfer Division and presented at the Semi-Annual Meeting, Los Angeles, Calif., June 14–17, 1943, of The American Society of Mechanical Engineers.

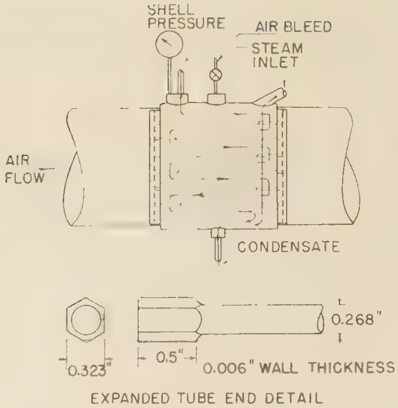

Fig. 2 Shell-Side Steam Flow and Expanded-Tube-End Detail

The steam employed as the heating medium in the shell was delivered to a separator at about 30 psig pressure. After liquid separation, the steam was throttled to about 3 psig to provide some superheat at the entrance to the core. The condensate from the core drained to a standpipe and then to an aftercooler where some subcooling was provided to minimize flashing. The condensate was metered gravimetrically.

The energy state of the superheated steam entering the core was established by pressure-temperature measurements and the steam tables. The condensate state was determined by temperature measurement upstream from the standpipe. Steam and condensate temperatures were measured by mercury-in-glass thermometers with the bulbs extending 3 in. in the fluid streams through glands.

Air-state measurements were taken as follows: Air humidity, by wet- and dry-bulb readings in the room; air stream entering the core; by a mercury-in-glass thermometer located 2 ft upstream from the core; air heating, by a calibrated shielded difference thermocouple (copper-constantan). The upstream junction had a stationary location at the previously mentioned mercury-in-glass thermometer. The hot junction was located 8 in. downstream from the core and was arranged to provide an 8-point vertical traverse of the air stream, with the traverse points located at the centers of equal annular areas. The arithmetic average of the temperature readings served to approximate closely the bulk air temperature inasmuch as the velocity distribution was quite uniform.

Air flow was metered by a central location of the pitot tube after an 8-point traverse calibration established the relation between the center-line velocity and the average velocity with respect to flow area, Fig. 3(*a*).

Fig. 2 reveals schematically the 7-pass baffle arrangement and the steam-flow path in the shell. Also shown is a detail of the expanded hexagonal ends of the round tubes.

Test Results

Typical approach-velocity-head distributions are presented in Fig. 3 together with the calibration for average to center-line velocity. Also shown are typical temperature distributions downstream from the core as measured by the air-heating difference couple. The arithmetic average of the 8-point temperature traverse readings was used to evaluate the bulk air temperature rise.

Table 2 summarizes the cold-core pressure-drop data. In this table the Reynolds number is evaluated employing a viscosity of 0.018 centipoises (air at 80 °F)

TABLE 2 COLD-CORE PRESSURE-DROP DATA

Air density ρ, lb/ft³	Tube-air mass velocity, G lb/(sec ft²)	Δp, in. H_2O	Friction modulus, $(f/2)(L/r_H)$	Tube Reynolds number[a] $N_{Re} = DG/\mu$
0.0740	1.19	0.105	0.90	2060
0.0740	1.54	0.153	0.80	2710
0.0740	2.28	0.290	0.69	4020
0.0740	2.94	0.430	0.62	5090
0.0743	3.52	0.585	0.59	6210
0.0740	3.40	0.550	0.59	6000
0.0745	3.87	0.685	0.57	6830
0.0740	3.72	0.635	0.57	6580
0.0742	4.38	0.89	0.575	7750
0.0741	4.23	0.83	0.575	7480
0.0741	4.98	1.15	0.575	8800
0.0745	5.66	1.465	0.57	10000
0.0740	6.32	1.85	0.575	11200
0.0745	6.38	1.855	0.57	11300
0.0737	7.75	2.75	0.57	13700
0.0735	12.7	7.05	0.54	22400

[a] Based on viscosity μ = 0.018 centipoises (80 F).

$$N_{Re} = \frac{DG}{\mu}\text{(dimensionless)}$$

which reduces to $N_{Re} = 1760G$ for G lb/(sec ft²). The friction-factor modulus is derived from the Fanning equation for the pressure drop

$$\frac{\Delta p}{\rho} = f\frac{L}{r_H}\cdot\frac{V^2}{2g} \text{ ft of air}$$

With G expressed in lb/(sec ft²) and p in in. of water this expression reduces to

$$\beta_f = \frac{f}{2}\cdot\frac{L}{r_H} = 167\frac{\Delta p}{G^2}\cdot\rho$$

The foregoing calculation assumes negligible air-density change in passage through the tubes, which is a valid idealization providing the pressure drop does not exceed 4 per cent of the total pressure and the air is not heated in the core.

Mechanical-energy dissipation, as expressed by the modulus given, includes entrance-shock loss, skin friction in the tubes, and exit-shock loss.

Heat-transfer and pressure-drop data, obtained with steam heating of the core, are summarized in Table 3. Energy balances for the greater part of the runs are within 5 per cent, attesting to the general accuracy of the temperature and flow measurements.

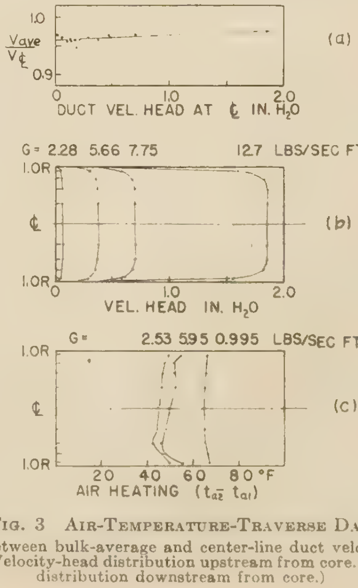

Fig. 3 Air-Temperature-Traverse Data

(*a* Relation between bulk-average and center-line duct velocities upstream from core. *b* Velocity-head distribution upstream from core. *c* Temperature distribution downstream from core.)

TABLE 3 HOT-CORE HEAT-TRANSFER AND PRESSURE-DROP DATA

Air density, ρ, lb per cu ft	0.0725	0.0725	0.0724	0.0720	0.0720	0.0720	0.0730	0.0725	0.0726	0.0720	0.0718
Tube-air mass velocity, G, lb/ft² per sec	0.995	1.57	2.53	4.15	4.48	4.67	5.38	5.95	7.62	10.1	13.5
Pressure drop, Δp, in. H_2O	....	0.220	0.435	0.945	0.98	1.19	1.47	1.78	3.1	5.3	9.5
Entering-air temperature, t_{a1}, F	87	86	85.5	88	89	88	84	86	83.5	91	84.5
Inlet temperature difference, $(t_s - t_{a1})$, F	133	134	134.5	130	131	132	136	134	136.5	129	136.5
Air heating $(t_{a2} - t_{a1})$, F	66.0	55.7	46.5	45.0	45.5	46.8	50.7	51.0	53.0	49.0	50.1
Number of transfer units, $\beta_q = (h/Gc)/(L/r_H)$	0.685	0.537	0.425	0.425	0.426	0.438	0.467	0.481	0.492	0.477	0.457
[a]Unit conductance, h, Btu/(hr ft² F)	4.2	5.3	6.8	11.0	11.9	12.8	15.8	17.9	23.4	30.0	38.4
[b]Reynolds number, $DG/\mu = N_{Re}$	1670	2620	4220	6920	7480	7800	8970	9930	12800	16900	22600
[c]Energy balance per cent difference	....	....	8	7	4.3	....	2.6	2.4	2.9	0.6	0.0

[a] Based on $\frac{L}{r_H} = 140$, $c = 0.242$ Btu/(lb °F).

[b] Viscosity μ for average air temperature $\frac{t_{a1} + t_{a2}}{2} = 110$ F, $\mu = 0.019$ centipoises.

[c] Expresses per cent difference between heat transfer to the air and heat transfer from heating steam, based on steam-energy rate.

The number of heat-transfer units $\beta_q = (h/Gc)(L/r_H)$ were evaluated from the test data employing the relation

$$\epsilon_a = \frac{(t_{a2} - t_{a1})}{(t_s - t_{a1})} = 1 - e^{-\beta_q}$$

The unit conductance h was then determined as

$$h = (\beta_q \, Gc)/(L/r_H)$$

This procedure is equivalent to employing the log-mean rate equation with a heat-transfer rate evaluated from the air-heating data.

Examination of the air-temperature-traverse data shown in Fig. 3 indicates that the air from the lower half of the core was not heated as much as the air from the upper half. Conductances based on the upper-half temperature data were about 5 per cent ($N_{Re} > 7000$) higher than the average conductances listed in Table 3. This condition suggests the presence of a scale resistance on the shell side. An attempt to clean the core with commercial solvents and acid was not entirely successful. The data reported in Table 3 are for the core after cleaning was attempted. If the conductance, based on the upper-half temperature data are typical for the clean core, the magnitude of the scale resistance included in the h and β_q data of Table 3, amounts to about $^1/_{500}$ (hr ft² deg F)/Btu.

The friction modulus $\beta_f = \frac{f}{2}\cdot\frac{L}{r_H}$ (as obtained from the cold-core runs) and the number of heat-transfer units β_q are related by the Reynolds analogy (2). The similarity of these two moduli is revealed in Fig. 4, where β_f and β_q are plotted versus N_{Re}. A consideration of this graph indicates that

$$\beta_f \cong K_f \beta_q \text{ for } 8000 < N_{Re} < 20{,}500 \qquad [1]$$

where

$$K_f = 1.2$$

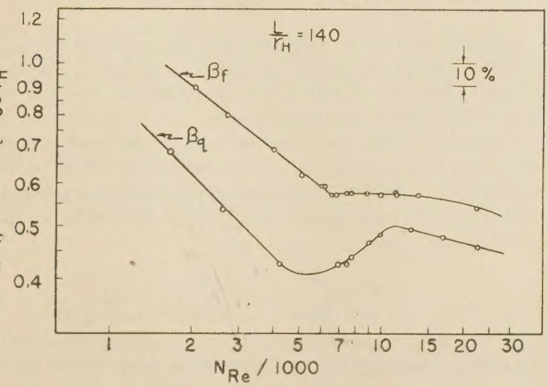

FIG. 4 EXPERIMENTAL HEAT-TRANSFER AND FRICTION DATA

The Reynolds analogy predicts a coefficient of 1 instead of 1.2 This increase can, in part, be attributed to shock at entrance and exit which contributes to the pressure drop, but is not associated with the skin-friction momentum and heat-transfer mechanism. Weise (3) reports a magnitude of 1.1 for the coefficient K_f.

An interesting revelation of these results (Fig. 4) is the definite dip in the characteristics starting at a Reynolds number of 10,000 for the β_q curve and 7000 for the friction-modulus characteristic. Normally, air flow with $N_{Re} > 4000$ is definitely turbulent in character, but for the core and test arrangement reported here fully turbulent flow obtained only for $N_{Re} > 10{,}000$. The reasons for this surprising result are not clear. Tentatively, it may be partially attributed to the following characteristics of the core- and test-duct arrangement:

1 Although the duct Reynolds number is 18 times that of the tubes (180,000 for a tube $N_{Re} = 10{,}000$), turbulence characteristic of this high Reynolds number was not established for the air entering the tubes because of the short approach section of the duct (4 diam), and the fact that the air was induced into the duct through a conical approach section from the relatively quiescent room conditions (Fig. 1).

2 The ends of the core tubes are expanded into hexagonals (Fig. 2) and thus provide a relatively smooth convergence from duct to tube flow. This minimization of shock at entrance may serve to perpetuate the duct-turbulence characteristic which is apparently considerably closer to laminar flow than the duct or even the tube Reynolds number alone would suggest, for the reasons first mentioned.

This rationalization is in part supported by the data of Washington and Marks (4) for flow through rectangular ducts having hydraulic radii of magnitudes comparable to that of the oil-cooler tubes. Their test arrangement provided for flow of compressed air into a relatively large inlet chamber, then through a rounded approach section into the duct of relatively small cross-sectional area. The turbulence of the air in the inlet chamber probably corresponded to quiescent air conditions and the smooth convergence approach section would tend to perpetuate this condition well into the duct. As a consequence the heat-transfer grouping h/Gc for the $^1/_8$-in. duct ($r_H = 0.0625$ in., as compared to the oil-cooler tube $r_H = 0.0640$ in.) departs markedly from the turbulent-flow characteristic in the neighborhood of Reynolds number = 10,000 (Bibliography reference 4, Fig. 8), which confirms the effect noted for the oil-cooler tubes.

This nonturbulent character of the flow at high Reynolds number is of more than academic interest in that turbulence conditions of the cold-air flow into an aircraft heat exchanger may correspond more closely to laminar conditions than to the turbulence characterized by the Reynolds number of the tubes. Therefore, predictions of heat transfer and pressure drop on the basis of turbulent flow may depart significantly from actuality.

For the core tested (ID = 0.256 in.) a $N_{Re} = 10{,}000$ corresponds to a 2 in. of water hot-core pressure drop, but for

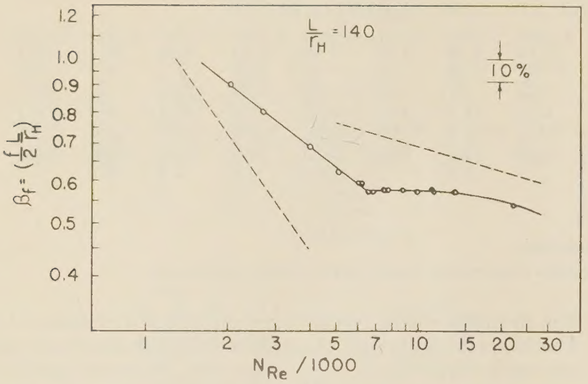

Fig. 5 Comparison of Predicted and Experimental Friction Data

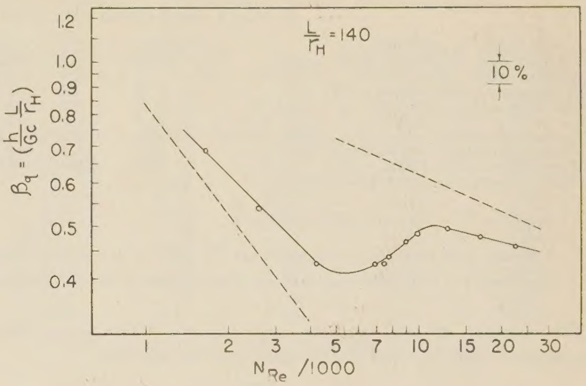

Fig. 6 Comparison of Predicted and Experimental Heat-Transfer Data

0.20-in-ID tubes of the same length at the same N_{Re} magnitude the pressure drop would be approximately 4 in. of water. Therefore this transition may occur in the design operating range of the oil cooler especially for the smaller tube sizes.

Fig. 5 compares the cold-core friction data with those predicted from available sources. The predicted curve is from the following equation given by McAdams (2)

$$\beta_f = \frac{f}{2}\cdot\frac{L}{r_H} = \frac{1}{2}\left[f'\frac{L}{r_H} + K_c + (1-\sigma)^2\right] \ldots\ldots\ldots [2]$$

The first term in brackets $f'\frac{L}{r_H}$ is the contribution of tube skin friction; the second term K_c accounts for the contraction shock at entrance; while the third term $(1-\sigma)^2$ is the contribution of the sudden expansion at exit from the tubes.

For $5000 < N_{Re} < 25{,}000$ (ref. 2)

$$f' = 0.046\ N_{Re}^{-0.2} \ldots\ldots\ldots [3]$$

and for purely viscous flow, $N_{Re} < 3000$

$$f' = 16/N_{Re} \ldots\ldots\ldots [3a]$$

A sudden contraction at entrance would introduce a K_c magnitude of 0.27 for the area ratio $\sigma = 0.53$. However, the contraction is not sudden as the expanded hexagonal tube ends produce a converging section $^1/_2$ in. long (Fig. 2). As a consequence $K_c \cong 0.27/2 \cong 0.135$ was assumed for this calculation.

These considerations introduced into Equation [1] result in the following relations used for the predicted curves in Fig. 5:

$$\beta_f = \frac{1}{2}\left(6.45\ N_{Re}^{-0.2} + 0.356\right) \ldots\ldots\ldots [2a]$$

for

$$5000 < N_{Re} < 25{,}000$$

and

$$\beta_f = \frac{1}{2}\left(\frac{2240}{N_{Re}} + 0.356\right) \ldots\ldots\ldots [2b]$$

for

$$N_{Re} < 3000$$

The exit and entrance losses contribute 20 to 30 per cent of the total friction.

Consideration of Fig. 5 reveals that the available friction-factor data adequately predict (within 15 per cent) the core behavior only for $10{,}000 < N_{Re} < 25{,}000$. For $N_{Re} < 3000$ the prediction based on viscous flow is not satisfactory since the entrance flow is not purely viscous and a parabolic velocity distribution does not obtain.

The comparison of experimental and predicted number of transfer-units data $\beta_q = (h/Gc)(L/r_H)$ is revealed in Fig. 6. The predicted characteristic for $N_{Re} > 5000$ was derived from the Dittus and Boelter correlation (2)

$$\frac{hD}{k} = 0.023\ N_{Re}^{0.8} N_{Pr}^{0.4} \ldots\ldots\ldots [4]$$

which for air $N_{Pr} = 0.74$ reduces to

$$\frac{h}{Gc} = 0.0275\ N_{Re}^{-0.2} \ldots\ldots\ldots [4a]$$

Tests by Brown and Barlow (5) performed on radiators having hexagonal tubes are in substantial agreement with Equation [4a], the magnitude of the conductances being only 4 per cent higher (coefficient of 0.0285 instead of 0.0275 in Equation [4a]).

For the laminar-flow region $N_{Re} < 4000$ the prediction of Norris and Stried (6) was employed

$$\frac{h}{Gc}\cdot\frac{L}{r_H} = 1.615\left(\frac{r_H}{L}\ N_{Pr}N_{Re}\right)^{-2/3} \ldots\ldots\ldots [5]$$

or for $N_{Pr} = 0.74$

$$\frac{h}{Gc}\cdot\frac{L}{r_H} = 1.97\left(\frac{r_H}{L}\ N_{Re}\right)^{-2/3} \ldots\ldots\ldots [5a]$$

Examination of Fig. 6 reveals that the predicted characteristic is within 20 per cent of the test performance for $10{,}000 < N_{Re} < 25{,}000$ and for $N_{Re} < 3000$. For the dip region, however ($3000 < N_{Re} < 10{,}000$) the agreement is poor.

Cold-core friction data together with the hot-core conductance data may be employed to predict accurately the hot-core pressure drop by use of the following approximate expression derived from Bernoulli's equation by assuming air density to be a function of temperature only.

$$\Delta p\,\frac{\rho}{\rho_0} = \frac{G^2}{g\rho_0}\left[\beta_f + \frac{\epsilon_a\,\Delta t_1}{2T_{a1}}\,(2+\beta_f)\right] \ldots\ldots\ldots [6]$$

In this expression $\Delta p\,\frac{\rho}{\rho_0}$ is the pressure drop (lb per sq ft) corrected to standard air-density conditions ($\rho_0 = 0.0750$ lb per cu

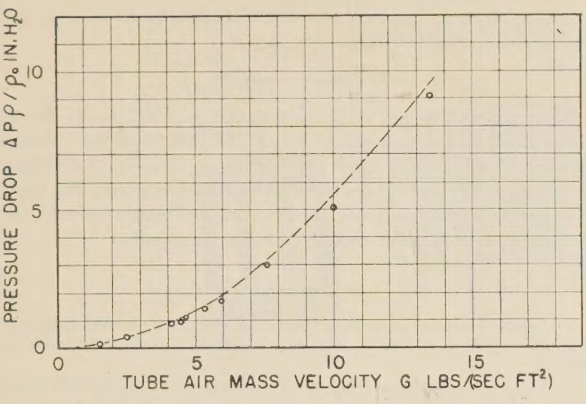

Fig. 7 Comparison of Predicted and Experimental Hot-Core Pressure Drop

ft). To convert to $\Delta p \frac{\rho}{\rho_0}$ in inches of water, divide by 5.20 (lb per sq ft)/(in. H_2O).

Hot-core pressure drop exceeds that of the cold core, for a given mass rate of flow, for the two following reasons:

1 Increased skin friction and exit-shock loss as a result of higher air velocity.

2 A pressure drop necessary to produce the flow acceleration.

In Equation [6] the term $\left(\frac{G^2}{g\rho_0}\right)\beta_f$ is the "cold-core" pressure drop while the term $\left(\frac{G^2}{g\rho_0}\right)\frac{\epsilon_a \Delta t_1}{2T_{a1}}\cdot\beta_f$ is the additional skin friction and exit loss arising from the higher velocity flow. The remaining term $\left(\frac{G^2}{g\rho_0}\right)\epsilon_a\frac{\Delta t_1}{T_{a1}}$ is the pressure-drop requirement for the accelerated flow. The hot-core pressure drop will exceed that of the cold-core drop, by 14 to 18 per cent.

Comparison of predicted and experimental hot-core pressure drop is illustrative of the applicability of Equation [6]. For this calculation, the hot-core temperature difference Δt_1 is the temperature difference, saturated steam to entering air $(t_s - t_{a1})$. Magnitudes of β_f and β_q were obtained from Fig. 4. Term ϵ_a is evaluated as $\epsilon_a = 1 - e.^{-\beta_q}$ The curve, Fig. 7, summarizes the results of these calculations and the data points of Table 3, corrected to standard density conditions, are superimposed for comparison. The agreement is within 5 per cent for most of the test points.

Oil-Cooler Performance

The basic air-side thermal-conductance and friction data may be employed to predict oil-cooler performance provided oil-side conductance data are available. Oil-cooling performance can be conveniently related to core pressure drop by plotting "oil-cooling effectiveness," ϵ_0 versus core pressure drop, $\Delta p \frac{\rho}{\rho_0}$ for a given oil rate, as in Fig. 8.

In terms of temperature conditions

$$\epsilon_0 = \frac{(t_{01} - t_{02})}{(t_{01} - t_{a1})}$$

That is, ϵ_0 compares the actual oil cooling $(t_{01} - t_{02})$ to the maximum possible cooling $(t_{01} - t_{a1})$, which would obtain were the core cross section very large. "Oil-cooling effectiveness" may also be expressed in terms of the over-all conductance, air-flow rate, and oil-flow rate by integration of the basic heat-transfer-rate relations and energy-balance expressions, assuming constant U along the tube. The resulting equation is

$$\epsilon_0 = 1 - e^{-\beta_0} \qquad [7]$$

where

$$\beta_0 = \frac{\epsilon_a G c_a A_c}{w c_0},\ \epsilon_a = 1 - e^{-\beta_a},\ \beta_a = \frac{U}{Gc_a}\cdot\frac{L}{r_H}$$

Over-all unit resistance $1/U$ may be established from the resistance concept as consisting of the summation of air-side, tube-wall, scale, and oil-film unit resistances. For this illustrative calculation, the tube-wall unit resistance is neglected as being small. An estimated scale resistance of $^1/_{500}$ (hr ft² °F)/Btu is included in the test magnitude of h (Fig. 4). Oil-film resistance for an oil-flow rate of 4000 lb per hr (at an average temperature of 200°F) is estimated as $^1/_{130}$ (hr ft² °F)/Btu (7). Thus, $\frac{1}{U} = \frac{1}{h} + \frac{1}{130}$. From this expression and Fig. 4, U was evaluated as a function of G (for an average $\mu = 0.019$ centipoises). This information together with the assumed oil-capacity rate

$$wc_0 = 4000\ \frac{\text{lb}}{\text{hr}} \times 0.49\ \frac{\text{Btu}}{\text{lb °F}}$$

substituted into Equation [7] permitted the evaluation of "oil-cooling effectiveness" as a function of air-mass velocity G.

The prediction of pressure drop was accomplished employing Equation [6], together with an assumed inlet-air temperature $T_{a1} = 460 + 60 = 520$ °R, and an average oil temperature of 200° F. Thus $\Delta t_1 = 200 - 60 = 140$ °F. Departures of 20 °F from these assumed temperatures will not materially effect the evaluation of $\Delta p \frac{\rho}{\rho_0}$. Air-friction-factor modulus β_f, for the core was determined as a function of G from Fig. 4 (for an average μ

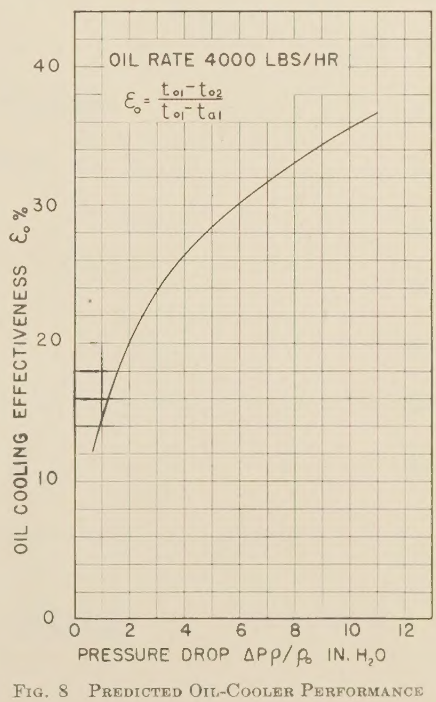

Fig. 8 Predicted Oil-Cooler Performance

= 0.019 centipoises). The foregoing procedure allows the determination of $\Delta p \frac{\rho}{\rho_0}$ as a function of air-mass velocity G. The results of the prediction calculation for the particular 9-in. core employed in the test work, are revealed in Fig. 8. A family of similar curves could be plotted employing oil rate as a parameter. Such curves, together with oil-side pressure-drop data, would provide a complete expression of the performance of an oil cooler.

Summary and Conclusions

1 Air-side thermal conductance and friction-factor data are presented for a particular 9-in. shell-and-tube-type oil-cooler core.

2 The estimated magnitude of the scale resistance (shell side) which is included in the reported magnitudes of the air-side film conductance amounts to about 1 per cent at N_{Re} = 1000 and 8 per cent at N_{Re} = 25,000. This effect, in part, accounts for the discrepancy between the predicted and experimental conductance data. It also suggests the necessity for allowing for this significant scale resistance in design calculations.

3 An unexpectedly early transition from turbulent to semi-viscous flow is observed as starting at N_{Re} = 10,000 for both friction and heat transfer. This behavior may possibly be a characteristic of the core-duct test system, but data are lacking to confirm this suggestion.

4 Because of this early transition, predictions from published heat-transfer and friction data are not adequate for $N_{Re} < 10{,}000$.

5 The cold-core friction data may be employed to predict hot-core pressure drop with adequate accuracy.

6 Air-side thermal conductance and friction data, together with oil-side conductance and friction data may be employed to evaluate oil-cooler performance with respect to oil-cooling and pressure-drop requirements.

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